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Modeling of High Capacity Passive Cooling System

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Program manager: Dr. Rengasmy Ponnappan

Principal Investigator: Dr. Md. Ashrfaul Islam
Professor
Department of Mech Engg
BUET, Dhaka-1000, Bangladesh

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14. ABSTRACT
High capacity passive cooling system studied in this project utilizes the thermoloop heat transfer concept (THTC). This device is an assemblage of Evaporator, Condenser, Non-return valves and Reservoir charged with a liquid for removing heat from any source upon which the evaporator is attached. Extensive experiments have been conducted and using experimental results a software module has been developed for to analyze the performance of the device for different operating conditions. Separate mathematical models for both evaporator and condenser are developed to evaluate heat transfer performance on the basis of thermodynamics and heat transfer. A software module has been developed to choose a suitable device for removing a specified heat at a particular temperature. The thermoloop device investigated in this project has the potential to be a better alternative for cooling electronic devices. Based on extensive experimentation, a software module has been developed to analyze the performance of the device. More extensive studies are necessary to mimic the operation of the device by a complete mathematical model.

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Content

Abstract

High capacity passive cooling system studied in this project is a thermoloop device that utilizes the thermoloop heat transfer concept (THTC). This device is an assemblage of Evaporator, Condenser, Non-return valves and Reservoir charged with a liquid for removing heat from any source upon which the evaporator is attached.

Extensive experiments are performed to with a view to understanding physics of the device by varying Evaporator volume, Condenser volume and Working fluids. Evaporator wall temperature fluctuations, fluctuation frequency (reciprocal of cycle time), condenser inlet and outlet temperatures are recorded during each experiment. It is found that working fluids having higher latent heat of vaporization have higher amplitude of temperature fluctuations with lower frequency. For example, Water has higher amplitude of temperature fluctuations with lower frequency compared with Methanol and Ethanol. Again, the higher the evaporator volume, the higher is the amplitude of fluctuation and the lower is the frequency. The temperature fluctuation with higher amplitude and lower frequency is always undesirable for electronic system cooling. Therefore, evaporator having smaller inner volume is better option for this device as far as temperature fluctuations are concerned.

Separate mathematical models for both evaporator and condenser are developed to evaluate heat transfer performance on the basis of thermodynamics and heat transfer. Estimating the flow rates of liquid and vapor, and the variation of system pressure correctly during the operation of the device are important challenges for the performance evaluation. Suitable instrumentation may help estimate these correctly in future.

A software module has been developed to choose a suitable device for removing a specified heat at a particular temperature. This module can simulate and analyze data for optimum performance of the device on the basis of experimental data taken so far.

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CHAPTER 1 INTRODUCTION

The major trend in the electronics industry in the recent years has been the development of faster and complex circuit technologies and higher packaging density. This drive towards more compact and faster technologies with shrinking system size has resulted in the tremendous increase of heat flux both at the chip and overall package levels. The significant evolution of the microprocessors has also resulted in the increase in the number of components to be integrated within the system and number of input/ output ports and other connectors. According to the International Technology Roadmap for Semiconductors, heat dissipation of high performance microprocessors is projected to approach the 160 W threshold within the next five years [1]. It also predicts that the resulting heat flux in certain regions of a microprocessor chip is projected to be over 100 W/cm^2. This rapid development in the electronics industry requires, for thermal management, high and efficient heat transfer in a small volume and space due to the closely packed microchips with higher power ratings.

Thermal management of electronic systems is one of the major focal points of the design system and is primarily concerned in keeping the temperature of the various components within a maximum allowable limit. It is argued that in the thermal management of electronic systems, the primary and secondary critical factors that the cooling solution must meet are the device's junction and solder temperatures. The choice of the cooling technique is thus determined not only by the power dissipation, but also by the junction temperature. A failure to maintain this temperature below the allowable limit results in the failure of the whole system. Therefore it is extremely important for effective thermal management of electronics system to precisely control the operating temperature of the critical components.

The most popular cooling devices for low heat dissipation systems today are the fans with heat sinks attached to the microprocessors, because of their low cost, ease of implementation, reliability and efficiency. Although the heat sinks are becoming more and more sophisticated in their design and coolant air velocities have increased significantly to meet heat dissipation requirements over the past decade, at certain point a limit will be encountered where nothing more can be done with a metal plate and fan.

But the close packaging of microchips and higher power ratings have increased the heat dissipation values to a critical level for high end electronics and computer processor cooling. The heat flux levels for the new designs of high density chips for the new generation of desktop computers has reached a value as high as 80 W/cm^2 [2]. From Fig.1.1 it can be seen that the projected chip heat flux can reach as high a value as 250 W/cm^2 in the year 2010 [3]. Again the board-to-board spacing has been decreased and consequently the available space for the thermal management of these devices has been reduced. Therefore, it is readily understood that a challenge of keeping the processor cool with a small size heat sink is becoming more and more difficult.

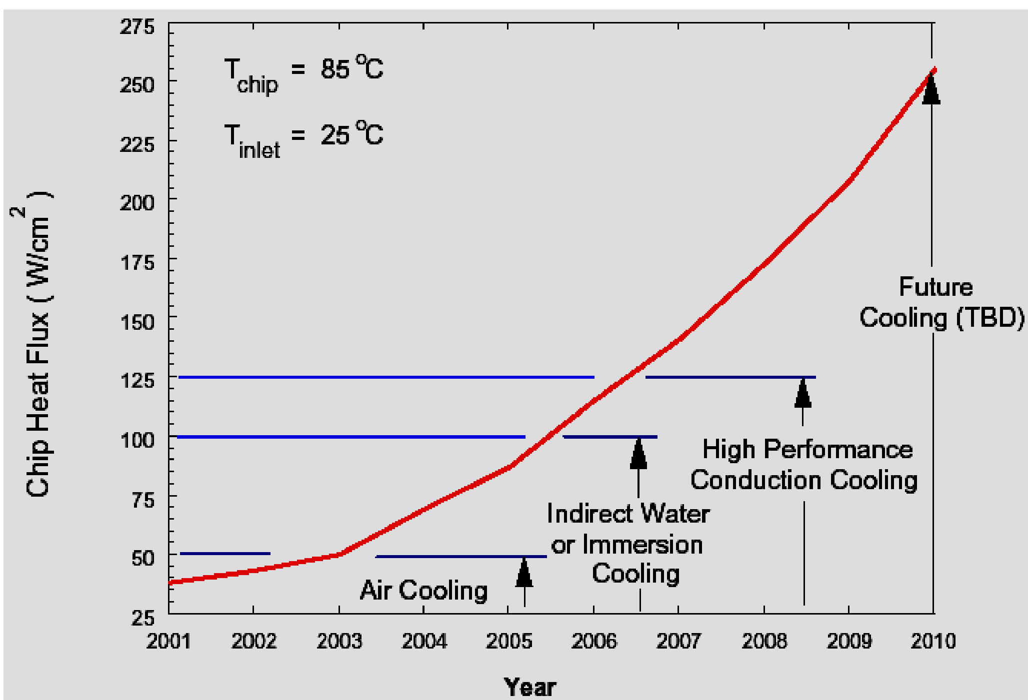


Figure 1.1: Projected Chip Heat Flux and Cooling Technology Limits (2002 thermal management roadmap- National Electronic Manufacturing Initiative – NEMI).

Therefore, new method of computer CPU cooling must be introduced with the capacity to dissipate as much as 200 W/cm^2 while keeping the temperature below 100^{0}C (preferably below 85^{0}C) and also ensuring very high uniformity of temperature over the surface [4]. Although there is no straight and simple answer to the question that whether the air cooling limit is reached, but it is fair to say that the currently available heat sinks are unable to meet the requirements and hampering the development of new generation computers. Therefore, searches are being conducted for highly efficient thermal management alternatives to handle this hugely multiplied heat loads of next generation electronic systems.

Thermoloop Heat Transfer Technology (THTT), also known as Pulsated Two-Phase Thermosyphon (PTPT) is one of the most recent technologies with the promise to be capable of dealing with very high heat flux requirements for electronics cooling and other high heat flux application [5]. It is very simple, passive and highly efficient operating against gravity with natural circulation and is capable to overcome the performance limitations of the indirect liquid cooling techniques like heat pipes and thermosyphons.

1.1 Motivation and Background

Today considerable interest and significant investment is being made on the search for a new, efficient cooling technology due to the increasing heat dissipation issue from the high end electronic systems. Future processors for high performance computers and servers have been projected to dissipate higher power in the range of 100–200 W/cm^2. The waste heat can accumulate and generate unacceptably high temperatures and thermal stress on the chip or in the package, resulting in reliability performance degradation and system malfunction. It is already an acknowledged fact that conventional air cooling techniques are about to reach their limit for cooling of high-end electronics. Experiments have shown that a maximum heat transfer coefficient of 150 W/m^{2}K can be reached with an acceptable level of noise, by using the standard fans [6]. This is equivalent to about 1W/cm^2 for a 60^{0}C temperature difference. According to the various researches carried out by the leading electronics companies of the world like IBM Corporation and Philips, these limitations of the available standard techniques are mainly due to the limited thermal conductivity of air for convection and copper for conduction [7].

Again the limit of heat dissipation of air cooling by parallel plate fin heat sinks for a desktop computer application was reported by Saini and Webb [8] for a 16×16 mm heat source at a temperature difference of 35^0C between the inlet air and heat source. It was reported that the maximum allowable heat load to be 95 W for a maximum chip heat flux of 37.2 W/cm^2. This can be increased by using a heat sink of greater size or increasing the air flow rate. But this would essentially increase the volume of package which is completely undesirable in this recent trend of miniaturization.

The comparison of heat transfer coefficient attainable for various electronic cooling techniques reveals that much higher heat transfer coefficients can be achieved by phase change liquid cooling than the conventional air cooling techniques. Therefore we can come into a simple conclusion that the future thermal management of electronic application will be based on liquid cooling techniques.

Because of their higher thermal conductivity and higher thermal capacity, liquids are significantly better heat transfer media than air. While air cooling technique is still used as the major means for cooling CMOS circuit technology, it is really being pushed to the limit. From Fig. 1.2 it can be seen that due to the demand of increased packaging density and speed, module heat flux level of recent CMOS circuit technologies has started to increase dramatically from the last decade [7]. Figure 1.2 also shows that the module level heat flux for CMOS technologies is now at the same level as the bipolar technology in the early 90's. But while all the bipolar systems employed some form of liquid cooling, most of the CMOS technologies continue to use air cooling schemes.

But as there is scope of marginal improvement with air cooling, many computer manufacturers are now taking course of liquid cooling. The Institute of Microelectronics (IME) predicts that existing air cooling techniques are reaching their limits due to the unfavorable thermo-physical properties of air [7]. Liquid cooling technologies, on the other hand, greatly enhance the cooling capacity, as coolant liquids possess larger heat capacities, high thermal conductivities and higher energy efficiency [9]. Even though liquid cooling can be employed with or without boiling, boiling can greatly reduce the electronics chip temperature compared with single-phase liquid cooling.

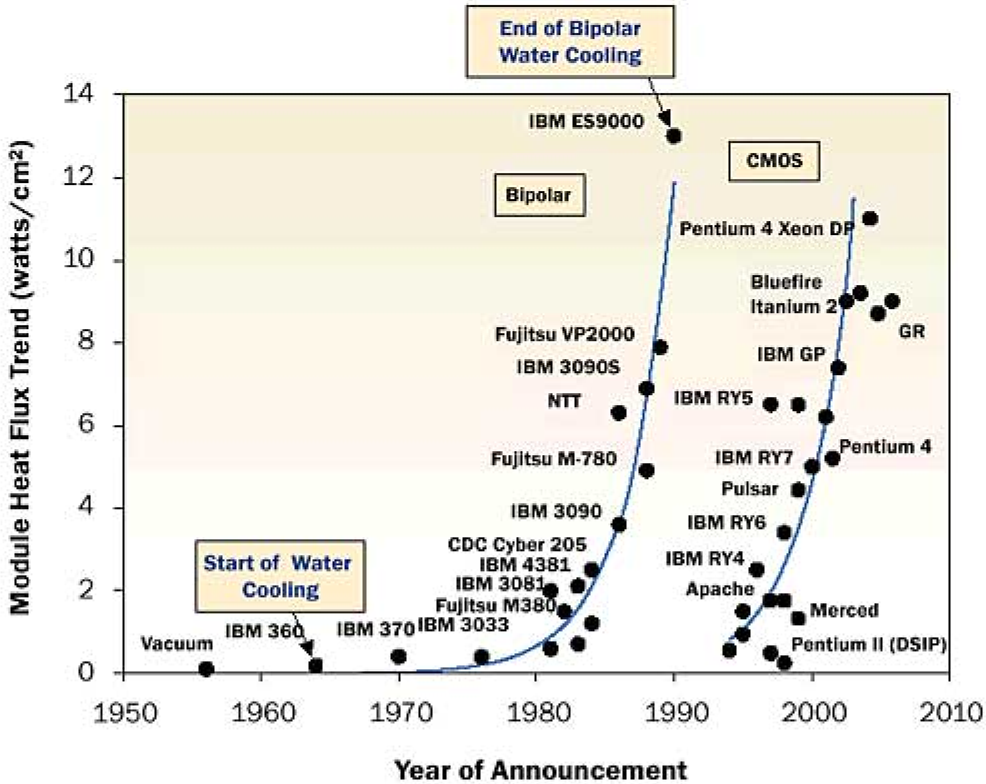


Figure 1.2: Evolution of module level heat flux in high-end computers [Source: IBM]

Liquid cooling techniques employed for cooling of electronic components can be classified into two broad categories- **Direct Liquid Cooling and Indirect Liquid Cooling.**

1.1.1 Direct liquid cooling

Direct liquid cooling can also be termed as direct liquid immersion cooling, because the liquid coolant comes into the direct contact of the components to be cooled. There is no wall separating the electronic chip and the surface of the substrate of the liquid coolant. High heat removal rate can be achieved by this method as the heat can be removed directly from the chip. Direct liquid immersion cooling has the advantage of greater uniformity of chip temperature than possible by air cooling.

The liquid immersion cooling technique has been used in microwave tubes in 1940s and also used later in cooling of high performance supercomputers [10]. But the implementation of direct immersion cooling in mainframe computers in the 1980s caused significant design complexities and associated increase in costs, which limits their prospect as a heat transfer technique for today's high density miniature electronics. One of the

disadvantages associated with direct immersion cooling is that the coolant has to be compatible with the device. Water is the most effective coolant but in most cases water can not be used as the direct immersion cooling liquid due to its electrical and chemical properties. Impinging jets, droplets and sprays are attractive direct liquid cooling options for removing high heat fluxes because of their associated heat transfer coefficients .These conventional direct liquid cooling techniques are not equally comparable to the performance of indirect liquid cooling schemes as shown in Fig. 1.3. Only direct water jet impingement appears to be a better performer than the best indirect cooling available.

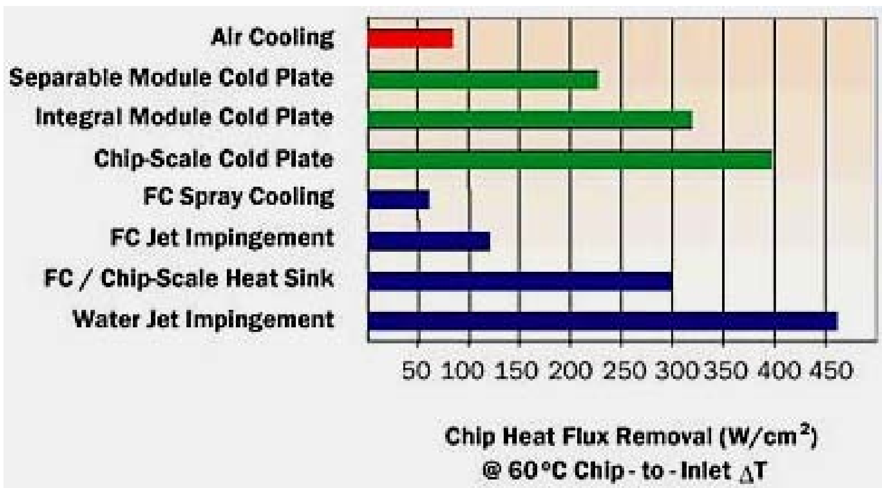


Figure 1.3: The heat removal capacity of different cooling techniques [10]

1.1.2 Indirect liquid cooling

Indirect liquid cooling technique is one in which the liquid does not directly contact the component to be cooled. In this method, a good thermal conduction is created between the microelectronic heat source and liquid cooled cold-plate attached to the module surface. Thermal interface materials (TIMs) can be used to increase this thermal conduction between the chip and the plate. In this method, water can be used as the working fluid to take the advantage of its superior thermo-physical properties, as there is no direct contact with the components.

The indirect liquid cooling techniques of electronics are drawing more attention of the researcher's to avoid the design complexities associated with the direct liquid cooling. Indirect liquid cooling is capable of removing heat at a rate two to four times than that of

air cooling [9]. The indirect liquid cooling techniques can be either Single Phase Forced Convection such as liquid cooled Cold Plates or Two- phase Forced convection such as Heat pipes and Thermosyphons.

Two-phase indirect heat transfer is a very appealing liquid cooling process as high heat flux removal can be realized through vaporization of the fluid in an evaporator attached to the heat source such as the microprocessor in computers. In the recent past, a number of electronic cooling schemes have been developed based on the two-phase heat transfer. Two-phase heat transfer, involving evaporation of a liquid in a hot region and condensation of vapor in a cold region, can provide the removal of much higher heat fluxes than can be achieved through conventional forced air-cooling. This is the reason why considerable researches are being conducted towards these approaches for thermal management of electronics. The most common forms of indirect phase change liquid cooling are the different types of- **Heat pipe and Loop thermosyphon.**

1.1.2.1 Heat Pipes

Heat pipes provide an indirect and passive means of heat transfer by liquid cooling. It is a simple device that can transfer heat from one location to another with very high thermal conductivity and often termed as ‘superconductors’ because of their excellent heat transfer capacity. Although the idea was first suggested in 1942 by R.S. Gaugler, it was in 1963 when G. Grover demonstrated the first heat pipe [2].

Heat pipes are sealed aluminum or copper containers which are partially filled with a liquid. The internal walls of the pipes and are lined with a porous medium (the wick) that acts as a passive capillary pump. When heat is applied to one side of the pipe the liquid starts evaporating. A pressure gradient exists causing the vapor to flow to the cooler regions. The vapor condenses at the cooler regions and is transported back by the wick structure, thereby closing the loop. The wick provides the capillary driving force by which the condensate is returned to the evaporator. A simple water-copper heat pipe will, on average, have a heat transfer capacity of 100 W/cm^2 [6]. Heat pipes are mainly used for cooling of electronic equipments such as laptop computers and in cryogenics and space technology.

Loop Heat Pipes: Loop Heat Pipes (LHPs) are particular kind of heat pipe where the condenser and the evaporator are separated and the working fluid is transported between them via tubings. In LHPs, unlike typical heat pipes, the wick structure exist only in the evaporator section [11]. LHPs possess all the advantages of the conventional heat pipes and provide reliable operation over long distance at any orientation. These devices can be considered as one of the most promising thermal control technologies for ground based applications as well as space applications. At present, different designs of LHPs ranging from powerful large size LHPs to miniature LHPs (micro loop heat pipe) have been developed and successfully employed in a wide sphere of applications. The most common working fluids used in LHPs are anhydrous ammonia and propylene. But other options, such as acetone and methanol which represents fewer hazards during manipulation and reduced distillation costs have also been used.

Pulsating Heat Pipes: Closed Loop Pulsating Heat Pipes, which is also known as Meandering Capillary Tube Heat Pipe or Closed Loop Oscillating Heat Pipe, has emerged in the recent years as a new electronics cooling technology. The Pulsating Heat Pipe is an innovating technology that has gained attention in the last 5 years [2]. Through the movement and phase changes in vapor bubbles and liquid slugs, heat is transferred from the evaporator to the condenser, as shown in Fig. 1.4**.** Higher operating temperatures are achieved when the PHP operates at the vertical orientation, while at horizontal orientation, the operating temperatures are lower. Pulsating heat pipes are capable of higher heat dissipation at relatively lower temperature difference between the evaporator and condenser [12]. The working principle and characteristics of LHPs and PHPs are discussed in detail in chapter 2.

Heat pipes are excellent heat transfer devices but their application is mainly confined to transfer relatively small heat loads over short distances. This limitation of heat pipes is mainly due to the major pressure losses associated with the liquid flow through the porous structure, called the entrainment losses. For the applications involving transfer of large heat loads over long distances, the thermal performance of the heat pipes is badly affected by increase in these losses. For the same reason, conventional heat pipes are very sensitive to the change in orientation in gravitational field. For the unfavourable slopes in evaporator-above-condenser configuration, the pressure losses due to the mass forces in

gravity field adds to the total pressure losses and further affect the efficiency of the heat transfer process. Thus the performance of the heat pipe may be insufficient to meet the continuous need for higher heat removal capacities over longer distance between the processor and the ultimate heat removal location. In addition, the high cost associated with heat pipes in comparison with the conventional forced air cooling places a limit on their use commercially.

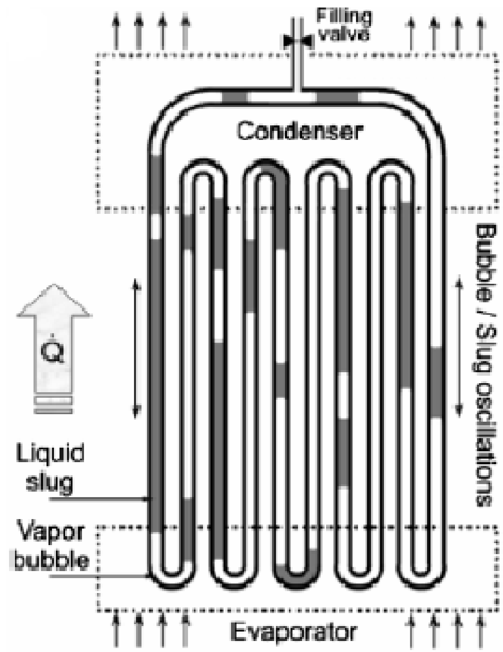


Figure: 1.4 Schematic of closed loop pulsating heat pipe

1.1.2.2 Loop Thermosyphon

Loop thermosyphon is regarded as a very promising solution for the high end electronics cooling because of its ability to meet the requirement of very high heat flux dissipation and also because of their promise to represent a low cost solution. A simple thermosyphon is a hermetically sealed chamber which is partially filled with a volatile working fluid. A simple thermosyphon differs from a heat pipe because condensate is returned to the evaporator of a thermosyphon by the pressure difference due to the difference in height rather than by capillary force as for a heat pipe. The driving force (pressure) must be greater than the sum of the pressure losses due to wall friction experienced by both vapor and liquid in the loop and other minor losses in fitting, curves etc.

A thermosyphon successfully implements two-phase liquid cooling by indirect contact with electronics. A two-phase thermosyphon basically consists of an evaporator and a condenser, which are connected through a passage, or a loop as shown in Fig. 1.5. The tube carrying the vapor from the evaporator is called rising tube or forward tube and the other tube connecting the condenser and evaporator is called the return tube or the falling tube. The fluid vaporizes in the evaporator as heat is transferred from the source to the evaporator. The vapor then moves to the condenser through the tubing where it condenses. The released heat is dissipated into the ambient from the condenser and the condensed liquid is returned to the evaporator, thus completing a loop. The density difference between the liquid and vapor creates a pressure head, which drives the flow through the loop and no other driving force is needed. Thus the major limitation of miniature electronic applications cooled by loop thermosyphons is that they are not capable to heat transfer and mass downwards, i.e. their performance is dependent on orientation.

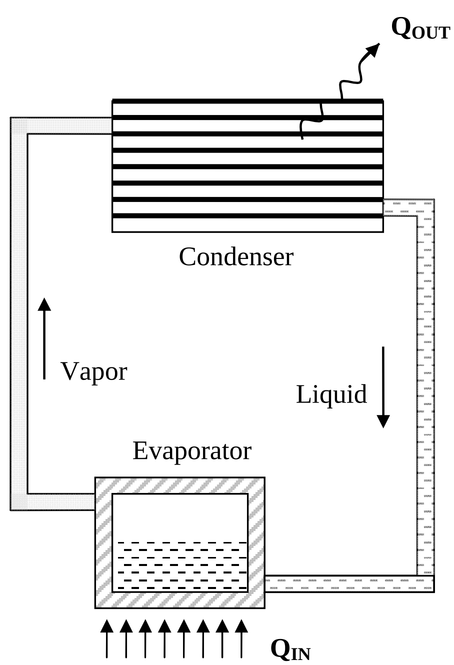


Figure 1.5: Schematic of an Indirect Thermosyphon Loop

The advantages of thermosyphon are as follows:

- Thermosyphon is a passive heat transfer device and it has no mechanical parts, which significantly improves its reliability issues. No energy is added to circulate the fluid and transfer the heat to the condenser by virtue of a pump.

- There is no geometric constraint on the shape of the evaporator. In some applications, this reduces the conduction resistances in transferring heat to the working fluid.
- There is no capillary limit in thermosyphons impeding the return of condensate as in the case of heat pipe. Thermosyphons can, therefore, move heat large distances and dissipate it efficiently unlike the heat pipes.
- Its quiet operation, in addition to the reduction in noise by replacing the heat sink fan, is a positive step for consumer and office desktop systems.
- It has flexibility in design and integration. The connection from evaporator to the condenser can be quite flexible, allowing a thermosyphon of a given design to function in many geometric configurations.
- When compared with liquid phase pumped cooling technologies, a thermosyphon has the advantages of simplicity, small size and low cost.

1.2 Pulsated Two Phase Thermosyphon (PTPT)

A Pulsated Two-Phase Thermosyphon (PTPT) named as 'Thermoloop' by Alam Thermal Solution Inc. [5] is a very recent technology with the promise of dealing efficiently with a very high heat flux requirements for electronics cooling and other high heat flux applications**.** It is a very simple and highly efficient technology and its operation is independent of gravity. It has the capability to overcome the performance limitations of the indirect liquid cooling techniques like heat pipes and simple loop thermosyphons.

The Thermoloop or PTPT consists of an evaporator connected to the heat source, a condenser which can be placed anywhere respect to gravity and a flexible or adjustable reservoir, separated from the evaporator as shown in Fig. 1.6. The evaporator is connected to the condenser through a vapor line and to the reservoir through a liquid return line. There are two check valves that control the flow in a fixed direction. The line that transports the vapor from the evaporator to the condenser is called 'the vapor line' or 'the forward line' and the line which brings back the liquid from the condenser to the evaporator through the reservoir is known as 'the liquid line' or 'the return line'. One check valve is placed in the return line between the condenser and the reservoir. The other check valve is inserted also in the return line between the reservoir and the evaporator.

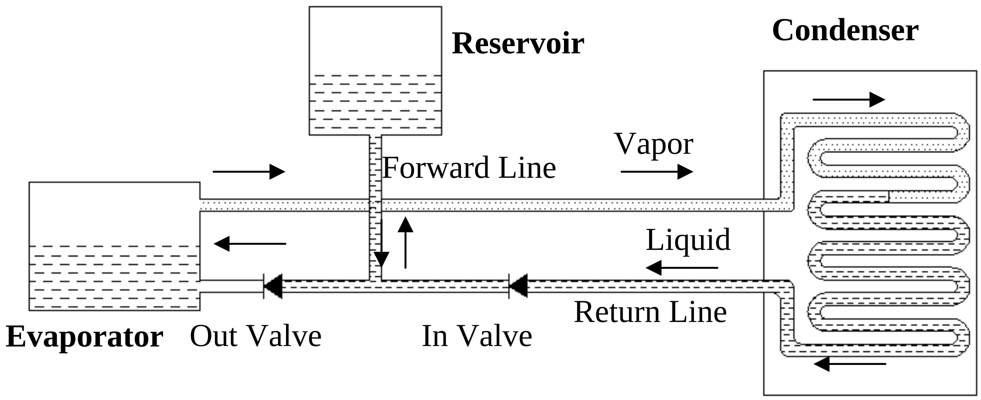


Figure 1.6: Simple schematic diagram of Thermoloop

Unlike the heat pipes like LHPs, there is no wick structure in the thermoloop. The important features of thermoloop are as follows

- Thermoloop can function against gravity where the circulation of the working fluid is forced without ant external work, but by the oscillation of pressure.
- Thermoloops are very similar in working principle with loop heat pipes and pulsating heat pipes.
- It has the capability to overcome high pressure drops with simple technologies, thus is capable to transport heat over a longer distance than heat pipes.
- As the Thermoloop device is very compact, can dissipate heat over longer distances and has a low cost of realization, it can be a very good choice for computer cooling and future electronic cooling demand.
- The device has no complex parts or manufacturing processes involved and can be made of cheap materials.
- It can be applied for wide range of cooling application ranging from cooling semiconductors, air conditioning to thermal management of outer space components.

When heat is supplied to the evaporator partially or fully filled with the working fluid, the temperature of the liquid starts to rise and when it exceeds the boiling point, the liquid starts evaporating and the pressure inside the evaporator increases. The device then starts operating in a periodic fashion by repeating two states: (a) vapor transfer state and (b) condensate return state.

(a) Vapor Transfer State: The liquid in the forward line and the condenser does not move forward until the pressure inside the evaporator is higher than the condenser. This state begins when the liquid inside the evaporator starts evaporating and the evaporator pressure starts to increase. The liquid in the forward line is pushed by the elevated pressure towards the condenser and then from the condenser to the reservoir where it is stored. The vapor can not move through the return line because it is closed by the check valve. Thus as the evaporator pressure increases, the condensed liquid moves into the adjustable reservoir, which accommodate the incoming liquid without increasing the system pressure. This process of pushing the liquid out of the condenser and forward line to the reservoir and replacing them by vapor from the evaporator continues as long as the rate of evaporation is higher than the rate of condensation. In this way the evaporator is gradually being empty while the reservoir is being filled and this continues until all the liquid inside the evaporator is vaporized. The liquid stored in the reservoir can not move back into the evaporator during this state because the one way out valve is closed as the pressure inside the evaporator is still.

(b) Condensate Return State: As only a fixed volume of the liquid is transported into the reservoir, the vapor transfer state ends and the liquid stored in the reservoir has to return to the evaporator to complete the cycle. This state is termed as the condensate return state. The pressure gradient between the reservoir and the evaporator must now change its sign so that the return of the liquid from the reservoir is possible. This means that either the pressure in the accumulator must increase or the pressure in the evaporator should decrease. In the thermoloop device, the pressure inside the evaporator decreases to the saturation pressure corresponding to condenser temperature as there is no liquid inside the evaporator.

The pressure in the evaporator decreases to a value lower than the pressure in the reservoir. As a result, the cold liquid from the reservoir enters into the evaporator as the out valve is now opened. In this way the cycle is completed. Thus, this device uses a two-phase fluid and operates in a periodic fashion with two sequential states of vapor transfer and condensate return and thus with periodic pulsation of pressure. This is why the device is termed as Pulsated Two Phase Thermosyphon (PTPT).

1.3 Brief Literature Review

The following are the conclusions of the Thermal Management Roadmap- published by the National Electronic Manufacturing Initiative (NEMI) in 2002 [3].

- Chip size may decrease with continued increase in circuit density resulting in higher heat flux.
- All new electronic products will most likely be air-cooled, including most computers, for the next few years.
- Portable (laptop) computers will need enhanced cooling technology in the near future despite the emphasis on low power dissipation.
- Cost will be a significant challenge for all future thermal designs and the speed to accomplish new designs will be vital to their success.
- High heat flux cooling capability is required for all high performance electronics.
- High thermal conductivity interface material is needed for heat sink applications.
- New cooling technology/system will be needed to handle increased heat load at product level.

According to this thermal management roadmap, significant improvement of the cooling technology will be needed in the area of indirect and direct liquid cooling along with several other techniques. From the point of view cooling of micro electronics components, two phase heat transfer devices seem to be the most suitable and promising scheme because of their high cooling capacity, greater energy efficiency, possible miniaturization of the device, lower maintenance requirement, safety, low cost and reliability. Different passive two-phase devices such as wicked heat pipes (flat heat pipe and loop heat pipe), wickless heat pipes such as pulsating heat pipes and wickless two-phase loop thermosyphons represent the most promising alternative for thermal control of electronic equipments. For this reason, a number of studies are being conducted to realize the applicability of these devices over the past few years.

1.3.1 Studies on LHPs AND PHPs

During the last decade, extensive researches have been carried out on a number of small scale prototypes of heat pipes on the applicability of those for electronics cooling. Loop heat pipes (LHPs) and pulsating heat pipes (PHPs) have been the most studied capillary driven devices with the ability to operate against gravity.

There has been an increasing interest in LHPs for cooling of high end electronics. The first actual application of LHPs in electronic cooling dates back to the end of seventies, when they were used for cooling of a unit of powerful transistors. Cooling of laptop computers had been the new sphere of LHPs application, owing to the development of miniature, compact, fairly efficient and cost effective devices [2]. The first experience in this direction came in 2001, when a number of compact coolers created on the basis of LHP dissipated a heat of the level 25-30 W from a laptop CPU [3]. LHPs for cooling of notebook computers are usually miniature size, with outer diameter from 2 to 4 mm for circular shape and thickness from 2 to 4 mm for flat rectangular shape. Copper tube and water filling are the best possible combination so far on the ground of compatibility and thermophysical properties [13]. Now copper-water LHPs with a flat oval evaporator 3.8 mm in thickness which can dissipate up to 100 W are being used largely in the laptop computers.

In the recent years, pulsating heat pipe (PHP), also called oscillation heat pipe (OHP) or meandering heat pipe, a particular wickless device capable of operating irrespective of gravity has emerged as a new cooling technology. Since the invention of PHP by H. Akachi in 1990, its operating principle and mechanism has been the subject of extensive research. The high heat flux removal capacity shown by some prototypes of PHPs has increased the interest in these devices. PHP has no complicated wick structure, is easy to produce and is capable of transporting higher heat rates at a relatively lower temperature difference between the evaporator and the condenser [14].

Yang et al. [15] reported an experimental study on the operational limitation of closed loop pulsating heat pipes (CLPHPs) for three operational orientations. They investigated the effects of inner diameter, operational orientation, filing ratio and heat input flux on thermal performance and performance limitation of CLPHPs. The CLPHPs were reported to obtain the best thermal performance and maximum performance limitation when they operated in the vertical bottom heat mode with 50% filling ratio.

1.3.2 Studies on Loop Thermosyphons

In spite of the excellent thermal performance shown by the different heat pipes, their relatively high cost in comparison with the conventional forced air cooling with heat sinks

place a limit on their use commercially. In this regard, concentration is increased on the development of two phase heat transfer devices without any capillary structure such as two phase loop thermosyphons. In the recent years, there has been tremendous improvement in the designs of loop thermosyphon which has drawn considerable interest in them.

Gavotti and Polasek [16] have given a detailed outline of the applicability of loop thermosyphon for the thermal control of electronic equipments. They have shown that loop thermosyphons are able to dissipate heat fluxes from electronic equipments up to a maximum value of 70 W/cm^2, depending primarily on the choice of the working fluid.

Tuma and Mortazavi [17] reported the test results on the use of thermosyphons for electronics cooling for two different test conditions. They tested with an evaporator of 40×40×4 mm^3, made of copper and two different condensers having volume of 440 cm^3 and 220 cm^3 respectively. The peak power attained by the thermosyphons was 170 W for the 1st condenser (volume of 440 cm^3) with two fans of 2800 rpm and 190 W for condenser volume of 220 cm^3 and two fans of 4200 rpm. The results are very competitive with emerging forced convection technologies.

Krustalev [18] also reported the test results of using loop thermosyphons for electronics cooling. The tests were carried out for an evaporator and condenser of same dimension (65×90 mm) where the condenser was located about 60 cm above the evaporator. The tests were performed with methanol as the working fluid. He reported a maximum heat flux removal of 16 W/cm^2, when the temperature gradient between the evaporator and the condenser was 22^0 C. Similar works are also reported on the possibility of loop thermosyphon for electronics cooling by Palm and Tengblad [19].

A comprehensive experimental and numerical study on the performance of a compact thermosyphon for desktop computer cooling was reported by Pal et al. [4], in association with Maryland University and Hewlett Packard. They performed the actual implementation of a compact thermosyphon on a current microprocessor. Their work described the design, construction and performance assessment of a two-phase compact thermosyphon for a Hewlett Packard Vectra VL800 PC with a peak microprocessor power dissipation of 80 W. The thermosyphon for the Vectra included an evaporator of 3.2 x 3.2 x 2.9 cm and a condenser 2.6 x 8.2 x 7.5 cm. The condenser was cooled by the system fan of 9.2 x 9.2 cm. Using de-ionized water as the working fluid, the temperature of the

evaporator base plate bottom was measured at 57°C under a chip power of 85 W and a local ambient temperature of 23°C. Since the specification called for a maximum case temperature of the CPU chip below 70°C, the thermosyphon was considered to be a great success.

1.3.3 Studies on the Choice of Working Fluid

One of the advantages of using a thermosyphon loop than immersion boiling is that the fluid may be chosen more freely as the liquid is not in direct contact with the components during normal operation. One may thus chose a fluid which needs small diameters of tubing and which gives low temperature differences in boiling and condensation and allows high heat fluxes in the evaporator. A thermosyphon may also be hermetically and permanently sealed which reduces the risk of leakage and allows the use of fluids with higher vapor pressures.

Palm et al. [20] has reported the results of the influence of the choice of working fluid on the design and performance of loop thermosyphon. They also reported how these goals are fulfilled for the case of closed external two phase thermosyphon loops. They found that high-pressure fluids gives better performance and more compact designs as high-pressure results in higher boiling heat transfer coefficients and smaller necessary tube diameter. Apart from the properties related to the performance of the loop thermosyphon (pressure drop, CHF, heat transfer coefficients of boiling, condensation heat transfer etc.), there are several other requirements which should be met, as identified by Palm et al. are:

- The fluid should not be harmful to people during production, normal operation or in case of a breakdown (sudden leak, fire, etc).
- It should not be harmful to the equipment in which it is installed. This means that it should not be explosive or flammable, not corrosive or otherwise incompatible with the materials of the equipment.
- It should not be harmful to the global environment. This means that it should have zero ozone depletion potential (ODP), it should not contribute to the greenhouse effect, not be hazardous to animals or plants or have decomposition products, which have such effects. Preferably, it should be a naturally occurring substance to eliminate the risk of unknown environmental effects.

- From an operational point of view, the fluid should be able to withstand the environment of the equipment for a long period of time without decomposing.
- Finally, it should have a low price and be readily available.

It is necessary to make some kind of compromise as no fluid meets all these requirements. This is done by considering various geometric and functional parameters of the system such as size, material, amount of heat to be transferred, application etc.

1.3.4 Studies on Thermoloop or PTPT

The major limitation associated with the possible application of loop thermosyphon for micro electronics cooling is that they are not gravity independent i.e. the condenser must be placed at a higher location with respect to evaporator. For this reason, some special two phase thermosyphon devices, operating in pulsating manner and capable of transporting heat and mass downwards were developed in the recent past, primarily for solar heating and cooling.

The idea of a Pulsated two phase loop thermosyphon, a particular pump-less heat transfer device operating against gravity without any capillary structure was first realized by Sasin et al. [22] in the Moscow Power and Engineering Institute. They, in fact, started this work with the idea of realizing some unsteady heat transfer device for the application of exploiting geothermal and solar energy [23]. Their invented device, operating on the principle of periodic pulsation of pressure and without any capillary structure, was able to transport heat and mass downwards with the condenser located below the evaporator.

Analytical studies of PTPT: Filippeschi [24] analyzed all the periodic two phase heat transport devices that operate against gravity and given these particular devices the generic name of Periodic two phase thermosyphon (PTPT) and reported a detailed analysis of the classification criteria of all the PTPT devices considering a large number of applications and their operating modes. The complex mechanism involved in the heat and mass transfer of PTPT devices was also explained using a mathematical code.

Experimental studies on PTPT: Fantozzi et al. [27] reported an experimental work on the upward and downward two phase heat transfer with miniature size PTPT. It presented an experimental study on the influence of the distance between the evaporator and condenser on the thermal resistance of the apparatus. Fantozzi et al. [28] presented a

feasibility study with a view to realizing a cooling device for desktop computer based on a PTPT heat transfer model. Using a computer code developed by the authors in cooperation with Moscow Power Institute and validated by the data collected in previous experimental tests, a mini experimental apparatus based on a PTPT heat transfer regime has been realized. The PTPT simulator code has been suited for the working fluid FC-72, a low boiling temperature dielectric fluid. The evaporator of the PTPT device is the simplest one with a flat exchange surface and a bottom heat regime of heat transfer. All the experiments are carried out filling the evaporator with 4×10^{-6} m^3 of working fluid FC-72 at the environmental condition (22 °C).

In 2006, Alam [5] came up with a design alternative to name it the thermoloop and presented a detailed analysis of the working principle, performance dependence and its comparison with the heat pipe [5]**.** He also reported some test results of thermoloop device for the cooling of high end computers. Alam reported that with the help of the two flow controllers and the adjustable reservoir, thermoloop device can produce much stronger vapor transport and condensate return forces, and thereby higher heat flux dissipation capacity. He also provided a conceptual sketch of the thermoloop device used inside a computer to provide thermal solutions for the microprocessor and other microchips.

1.1 Scope of the Present Work

High capacity passive cooling system studied in this project is a thermo-loop device that utilizes the Thermoloop Heat Transfer Concept THTC. This project involves two major development areas:

- Development of a mathematical model on the thermo-loop heat transfer concept (THTC) based on thermodynamics and heat transfer.
- Putting mathematical model into user friendly computer model in order to study the performance of the thermoloop device.

This model is to have two major modules: **Design Module–**Device components, materials and working fluids will be chosen on the basis of experimental results and better heat transfer performance and **Analysis Module–**Performance of the device at different thermal loads and ambient conditions to address the relationship among pressure limit, heat flux limit, operating temperature, fluctuation frequency, and fluid physical properties.

Experiments were carried out with 14 different prototypes: a combination of 6 different evaporators and 4 different condensers, of the device of varying dimension and materials of the evaporators and condensers and tested under different working fluids. Four different types of fluids, Ethanol, FC-72, Methanol and Water had been used to conduct experiments. Measurement of various parameters was carried out to characterize their effect on the overall heat transfer performance of the thermoloop device.

2.1 Experimental Setup

Various components of the thermoloop test setup were connected by different types of connectors and proper adhesive to maintain firm and leak-proof joints. Both natural and forced convection conditions were tested in the experiment. In forced convection, the condenser convection condition was varied by using a forced draft fan. Photographic views of the experimental setup, forced convection and natural convection, of the thermoloop device are shown in Figs. 2.1. and 2.2, respectively.

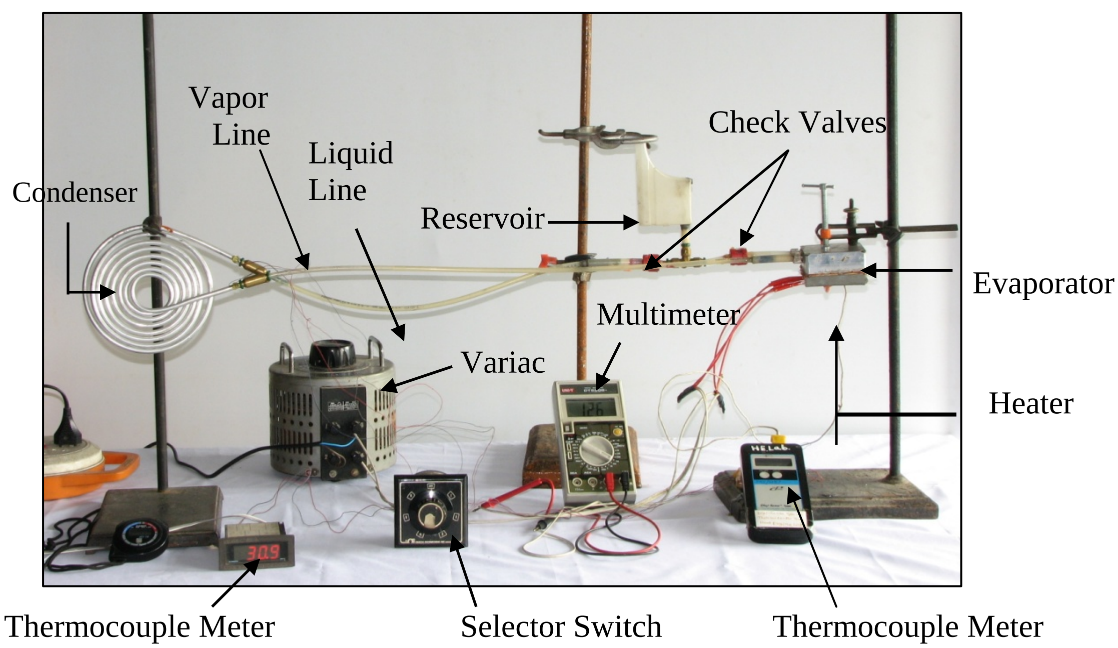


Figure 2.1: Front View of the experimental set-up (Forced convection)

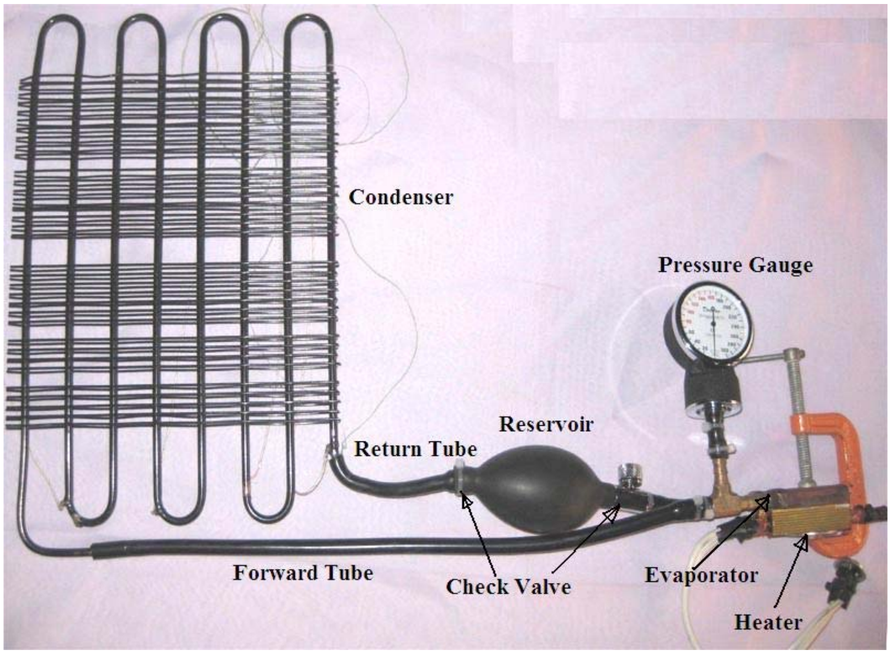
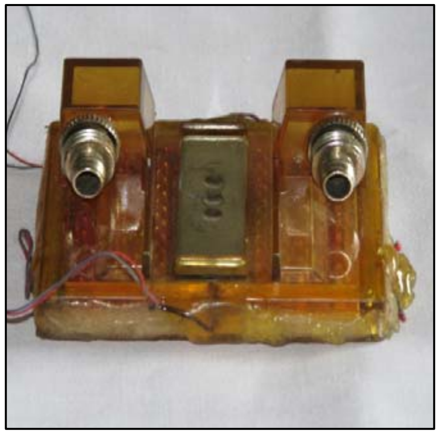


Figure 2.2: Front view of the experimental set-up (Natural convection)

As heat is applied to the evaporator by the heater connected to it, the temperature of the liquid stored in the evaporator increases. When the temperature exceeds the boiling point of the liquid, it evaporates and vapor bubble starts to form. Pressure increases and vapor pushes the liquid in the forward line towards the condenser, and then from the condenser to the reservoir. Flow can not occur in the opposite direction due to the two check valves placed on either side of the reservoir. Thus the liquid coming out of the condenser is stored in the reservoir as long as evaporation is continued in the evaporator. When the vacuum pressure inside the evaporator exceeds the pressure drop of return line and the non-return valve between the reservoir and the evaporator, the non-return valve opens and the liquid in the reservoir flows back into the evaporator. In this way, the cycle completes and a new repeating cycle starts.

2.1.1 Evaporator

Evaporator is the most important component of the thermoloop the device. In the present study, 6 prototypes of the evaporator of different dimensions were realized (Fig. 2.3). Here, E75 means an evaporator of inside volume 75 cc. All of the evaporators had a flat bottom surface for attaching the heaters (serving as heat source) conveniently. Internal surface of two large evaporators (E75 and E 25) were grooved to enhance the boiling heat transfer features (Fig. 2.4) and others were flat. All the evaporators are provided with suitable attachments to connect the forward and return lines easily and without any leakage.



(a) **E75** **(b) E25** **(c) E15**

(d) E07

(e) E03

Figure 2.3: Evaporators used in the thermo-loop devices

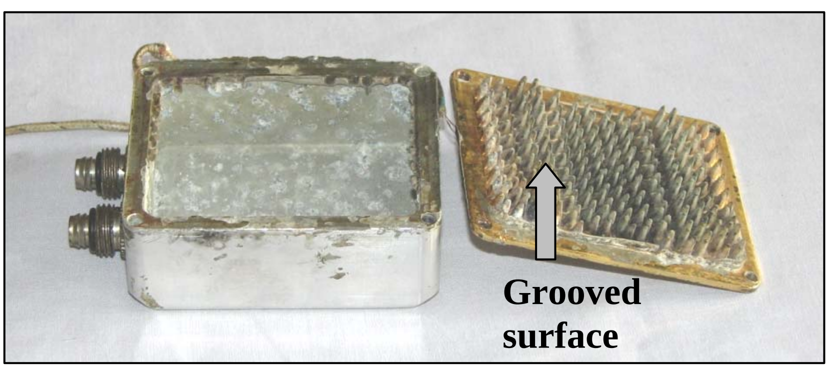


Figure 2.4: E75 with boiling enhancement inside the evaporator

Table 2.1: Geometric features of the evaporators

Dimension	E03	E07	E10	E15	E25	E75
Heat Input area (cm^2)	12	12	12	12	12	12
Inner volume (cc)	2	7	10	15	25	75
Material	Copper	Copper	Copper	Copper	Copper	Copper
Boiling surface	Plane	Plane	Plane	Plane	Enhanced	Enhanced

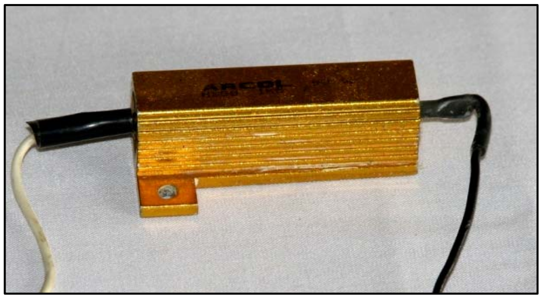
Table 2.2: Specification of the heaters

Features	Heater type-1	Heater type-2
Heating surface	Flat	Round
Length (mm)	50	55.6
Heating capacity (W)	50	150
Material	Copper	Aluminum
Number of heaters	2	2

The shape and dimension of all the evaporators were chosen to ensure that they are compact and are miniaturized for the practical application in electronics cooling. Considerable thought was given in their design to make measurement and instrumentation easier. Proper care was taken in perfectly sealing the evaporators to ensure that they can withstand the high pressure developed inside during the evaporation of the fluid. Gaskets were used for the two large evaporators to prevent leakage and consequent pressure drop during the boiling process. It will be shown later that the slightest leak in the evaporator can lead to drastic consequences. One thermocouple was connected to the evaporator to give the wall temperature of the evaporator. The dimension and other geometric features of the evaporators are given in Table 2.1.

To supply the required heat flux to the evaporator, two different types of AC heaters were employed in the current study. Both these heaters were resistive heating elements whose output can be controlled by connecting them to the electric line through a variac. The characteristics of these two kind of heaters employed are provided in Table 2.2. The flat heaters were made of copper and had a capacity of 50 W. For higher heat input, they were connected in parallel. The

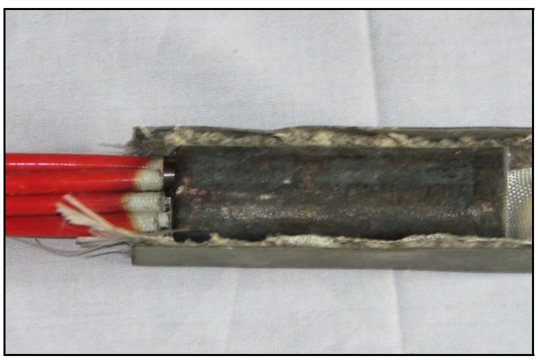
heaters of higher heating capacity (150 watt each) were of cylindrical shape and had a diameter of 8 mm each. As the evaporator surface is flat, these two round heaters were pressed into a close rectangular copper box (40×20×10 cm) with flat surface to facilitate the connection with the evaporator.



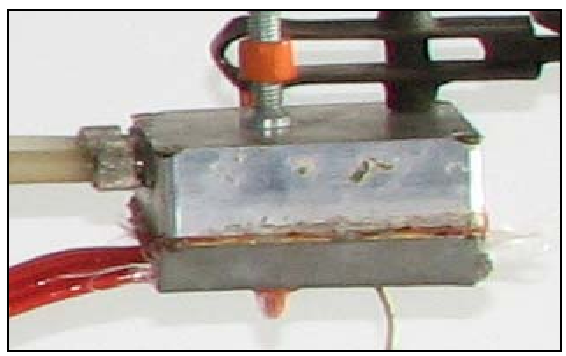
(a)

(b)

Figure 2.5: (a) Flat heater of 50 W capacities, (b) Cylindrical heaters of 150 W capacities pressed inside a copper box



(a)



(b)

(c)

Figure 2.6: (a) The cylindrical heater with the copper cover placed inside the rectangular mild steel box with insulation, (b) &(c) Heater- evaporator assembly

The heaters were connected to the evaporators by clamping [Fig. 2.6(b) &(c)]. To minimize the thermal contact resistance between the evaporator and the heater box, a thin layer of thermal interface material (TIM) were used between them. The thermal interface material used was Thermolite® which had a contact resistance less than 0.275 ^{0}C-cm^2/ W.

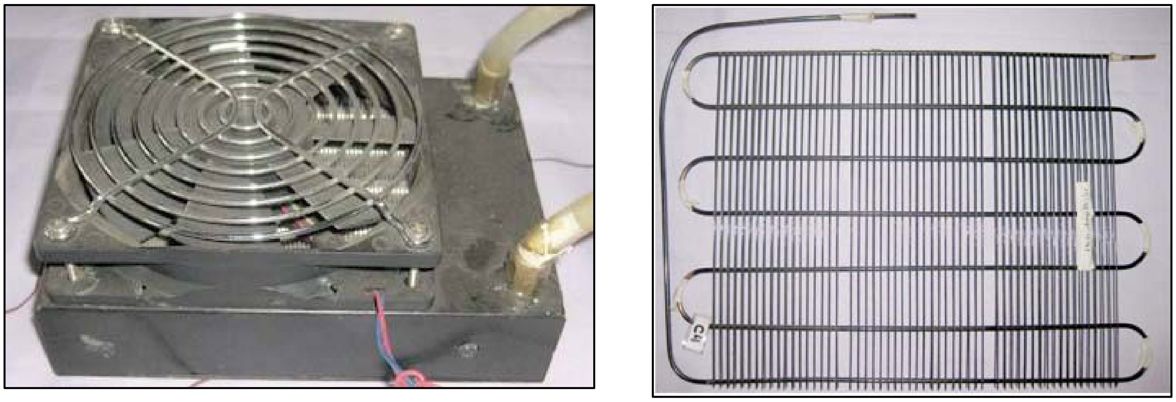
2.1.2 Condenser

Fantozzi et al. [28] have reported that the design of the condenser has a significant effect on the heat transfer characteristics of pulsating thermosyphons. They concluded from experimental results that, if the condenser is designed for a cooling power lower than the input power of the evaporator, it is possible to obtain a cooling of the reservoir and of environment, otherwise an environmental heating is realized.

In this experiment, 4 different types of condenser of different dimensions and different cooling conditions were utilized (Fig. 2.7). One condenser was attached with fan to facilitate forced convection and other three condensers used for natural convection and were different only in length of the condenser tube as shown in Table 2.3. The condensers were connected to the forward and return tubes making the joint firm and leak-free. In forced convection, two thermocouples were connected to the condenser to measure the inlet and outlet temperatures of fluid and in natural convection seven thermocouples were attached to different location of the condenser. In forced convection condition, to increase the rate of heat dissipation from the condenser, the convection condition of the condenser was changed to forced convection by employing a forced draft fan (Fig. 2.8).

Table 2.3: Geometric features of the 4 condensers employed

Dimension	C50	C35	C25	C20
Tube diameter, OD(mm)	6.25	4.5	4.5	4.5
Tube diameter, ID(mm)	4.5	3.8	3.8	3.8
Total length (mm)	2410	2800	2400	2200
Volume (cc)	50	35	25	20
Material	Copper	Copper	Copper	Copper
Number of Turns	8	8	6	4
Convection condition	Forced	Natural	Natural	Natural



(a) C50 **(b) C35**

Figure 2.7: Condenser used for (a) C50, (b) C35 the employment of Thermoloop

Figure 2.8: Fan used for forced convection

2.1.3 Forward and Return Lines

The forward and return tubes transport the vapor and liquid, respectively, between the evaporator and the condenser and hence form the most important loop of the thermosyphon. The tubing of the first prototype had the same outside diameter, thickness and length for the forward and return lines. The check valves and the reservoirs were inserted at certain distances into the return line (Table 2.4). All the joints in the forward and return lines were maintained firmly affixed to prevent any kind of leakage. The reservoir and the check valves (described in sections 2.1.4-2.1.5) were properly glued after inserting them into the return line. The flow of the liquid and vapor through the two lines was possible to observe visually because in both the tubing some portion were made of transparent poly-ethylene material, capable of withstanding high temperature and pressure. The other tubes were opaque and made of rubber to minimize heat loss.

Table 2.4: Geometric features of the forward and return tubes

Name	Dimension
Length of forward tube (mm)	200
Length of return tube (mm)	100
Outside diameter of forward tube (mm)	6.25
Outside diameter of return tube (mm)	6.25
Thickness of forward and return tubes (mm)	1.016
Length of return tube from reservoir to the evaporator (mm)	100

2.1.4 Reservoir

Fantozzi et al. [29] has pointed out that it is possible to control the temperature of the evaporator by correctly designing the reservoir's surface and its heat loss to the environment and its temperature. In this experiment, two type of reservoir were used. A rectangular reservoir (6×2.7×6.2 cm) with curved upper part and made of plastic was used to store the liquid being pushed out of the forward and return lines and the condenser. It had a small opening at the top so that the pressure inside the reservoir is always atmospheric when empty. The reservoir was fitted at a height of 2.5 cm and 5.5 cm above the return line for forced convection experiments. The total liquid storage capacity of the reservoir was about 80 cm^2. As the reservoir was transparent like the tubing, the accumulated liquid height in the reservoir was visible from outside. During the experiments, the reservoir was kept straight to measure the accumulated liquid column height which served as a measure of the system pressure. The another reservoir, used during natural convection experiments, was a plastic bag of 200 ml used to store the liquid being pushed out of the forward and return lines and the condenser. The reservoir was flexible and closed in order to maintain atmospheric pressure inside the reservoir. During the experiments, the reservoir was kept at the same level of condenser and evaporator so that effect of gravity can be ignored. The reservoir was not thermally insulated in the present study.

2.1.5 Flow Controllers

To maintain the flow of the working fluid in the desired direction, two similar kind of one-way check valves (non-return valves) were inserted into the return line on either side of the reservoir. The check valves were placed at a distance of 5 cm from the reservoir in each side. The check

valve placed between the reservoir and condenser was called the ‘In-Valve’. Its function was to permit the flow only from the condenser towards the reservoir. The other valve, placed between the reservoir and the evaporator was called out-valve and it allowed the flow only from the reservoir towards evaporator. The check valves were made of plastic and had the dimension of 2.5 cm in length and 2 cm in diameter and opening pressure of the non-return valve was 500 Pa.

2.1.6 Working Fluid

The properties, especially the boiling point and latent heat of vaporization of the working fluid determines the nature and extent of the heat transfer and the heat transfer performance of the device. The maximum temperature of the evaporator wall and the condenser inlet and outlet temperatures depend on the boiling point of the working fluid. The four working fluids used in these experiments study were water, ethanol, methanol and FC-72. These fluids were selected because of their excellent thermo-physical properties and wide availability. The most suitable fluid to be used for electronics component cooling by thermoloop is known to be different types of dielectric fluid such as FC-70 or FC-72 etc. FC-72 is an ideal fluid to use in low temperature heat transfer applications. Its high dielectric constant also means that it will not damage the electronic equipments in the case of a leakage or other possible failure. FC-72 liquid is chemically stable, compatible with sensitive materials, nonflammable and practically non-toxic. But it has low latent heat of vaporization which limits heat transfer capacity in some cases.

2.2 Data Acquisition

For experiments of forced convection a K-type thermocouple was connected to the evaporator wall and the temperature was measured by a DIGI-SENSE® K-type thermocouple thermometer (Model: 91000-00, Range: -50 to1200^0 C, Accuracy: ±0.5% of reading or ±1^{0}C**)** for some experiments. The T-type thermocouples connected to the condenser were connected to an Omega DP 115 TC-type thermocouple thermometer (model- DP 115, range -200 to 200 0C, accuracy: ±10C) via selector switch. For experiments of natural convection a Data logger (Pico Technology) of eight channels was used to measure temperature of eight location of the thermoloop device. The power supplied to the evaporator was changed by varying the input voltage with the help of a variac. The input voltage to the heater was measured using a DT 890BT digital multimeter**.** The amount of current drawn by the heater was measured by a HIOKI

2261 digital clamp on meter with a resolution of 0.01 ampere. The height of the liquid collected in the reservoir was measured using measuring tape. All these reading were taken simultaneously at a fixed interval of time.

2.3 Experimental Procedure

The experimental set-up (Shown in Figs. 2.1-2.2) consisting of thermoloop device and instrumentation for measuring temperature and pressure in different locations was mounted on a rigid structure in the heat transfer laboratory of Mechanical Engineering Department, Bangladesh University of Engineering and Technology. The evaporator and the condenser were attached firmly to the structure by clamping. For experiments of forced convection the reservoir was also held by a clamping frame to keep it upright, so that it does not tilt or get lowered by the weight of the accumulated liquid. And for natural convection setup reservoir was held maintaining the same level with evaporator and condenser.

The experiments were conducted using water, ethanol, methanol and FC-72 as the working fluids. Before every test, the device was filled with working fluid creating suction pressure at the outlet of the condenser, when the evaporator inlet was connected with fluid reservoir. Whole evaporator, forward and return lines and the condenser were filled with the fluid. Then the reservoir with some liquid inside was attached with the condenser outlet and evaporator inlet. Two non-return valves were connected at the two ends of reservoir to ensure fluid flow in the direction of condenser outlet to evaporator inlet through reservoir. Considerable care was taken to eliminate the formation of bubble or gas pockets in the evaporator, forward and return lines and also in the condenser. All the tests were performed by placing the thermoloop setup horizontally (i.e. keeping the evaporator and condenser and reservoir at the same level of elevation).

After all is ready, the power connection to the heater was switched on and the desired heat input was ensured by setting the variac to the required level and taking consequent current reading. After the power input is set, the temperature of the evaporator and condenser and the height of the accumulated working fluid, for forced convection, in the reservoir were recorded at a regular interval of 10 seconds for forced convection and 4 data were taken in a second for natural convection.

The total time required for completion of every heat transport cycle was recorded for all the tests performed, along with readings of maximum wall temperature of the evaporator, maximum inlet and outlet temperature of the condenser and maximum height of the liquid collected in the reservoir for every cycle of forced convection process. When experiments were performed for natural convection, temperatures were taken from eight different points of that device. As the forward and return lines were transparent in some portions, the motion of the liquid through these lines and its nature of oscillation due to pressure pulsation were also closely observed to understand the physics of the system. Almost all the experiments were carried out for minimum of 12-15 heat transport cycles. Experiments were stopped if the evaporator temperature exceeded 150^0C due to reasons such as leakage or evaporator dry out. During some tests, the experiments were continued for about four-five hours to observe the performance of the device in long period of operation.

2.4 Uncertainty Analysis

The benefits and importance of uncertainty analysis have become widely recognized nowadays. This is a very powerful tool of identifying the source of error in the experiments. The results of the uncertainty analysis performed on the primary measurands and calculated quantities in the present study are presented in Table 2.5 and Table 2.6 respectively as per procedure adopted by Kline and McClintock [36].

Table 2.5: Uncertainty in the measured values

Measurands	Precision limit, P	Bias limit, B	Total limit, W
Time (t)	0.25%	0.02%	0.25%
Temperature (T_E)	1.5%	0.5%	1.58
Temperature (T_C)	1.82%	0.5%	1.9%
Diameter (Di)	1.21%	0.94%	1.61%
Length (L)	0.71%	0.09%	0.71%
Width (b)	2.78%	1.29%	2.10%
Height (H)	1.72%	0.86%	1.92%
Area (A)	2.62%	1.87%	2.22%
Volume (V)	5.16%	2.58%	5.77%

Table 2.6: Uncertainty in the calculated quantities

Quantity	Total Uncertainty (U)
Volume flow rate (Q)	7.25%
Mass flow rate (m)	7.25%
Velocity (v)	6.61%
Pressure drop (ΔP_{CON})	10.0 %
Pressure drop (ΔP_{FL})	7.11%

CHAPTER 3 MATHEMATICAL MODEL

3.1 Background

A Pulsated Two-Phase Thermosyphon (PTPT) or thermoloop is very similar in working principle to a heat pipe as both are passive indirect cooling devices and in both the cases pressure is the driving force to move the vapor from the evaporator to the condenser. But thermoloop differs from a heat pipe in the manner the condensate is returned to the evaporator. In thermoloop, condensate is returned to the evaporator by suction due to vacuum pressure generated in condenser hence in evaporator than by capillary force as in a heat pipe. The driving force (pressure) must be greater than the sum of the pressure losses in the components, pressure losses due to wall friction experienced by both vapor and liquid in the loop and other minor losses in fittings, bends etc. For the devices that utilizes the capillary force as the driving force for the return of the liquid, the pressure and temperature variations are considerably low as the thermal resistances are very small. But for thermoloop, the pressure and temperature variations are much higher due to higher thermal resistances in the system.

In a PTPT, both the mass and heat flow depend on time as the device operates in a periodic manner. Sasin et al. [22] developed a time dependent mathematical model for the first time for PTPT device, which was based on the result of the experimental observation. They investigated the thermal behavior of the device by dividing the heat transfer cycle into four different parts. Then Fantozzi et al. [29] proposed another mathematical model based on the original model of Sasin et al. and it was verified by experimental results. They remodeled the original one of Sasin et al. and classified the heat transfer cycle into two distinct portions – a transfer operation from evaporator to reservoir and another return operation from reservoir to evaporator. They divided the whole system into seven control volumes to determine the energy and mass transfer rates. Lumped capacitance method was used to

determine the properties within each of the control volumes with several assumptions for the first heat transport cycle. They performed validation of their model by experimental data and reported good qualitative accordance, the error being less than ±2%.

In the present study, an analysis of the physics involved in the working principle of the thermoloop is investigated and a simplified mathematical model has been developed for the prediction of the thermal behavior of the device. From experimental observation the operating processes of thermoloop have been divided into four time zones, no flow time, slug flow time, vapor flow time and the liquid return time. Heat and mass balance has been performed on different components individually, and the pressure pulsation and the variation of operating temperature have been investigated mathematically. Several assumptions have been made to avoid the complexity of two- phase periodic heat transfer process.

3.2 Assumptions

Following assumptions have been made to facilitate the mathematical modeling of a two-phase periodic heat transfer process without unwarranted complexity.

i. At the initial condition of every cycle the thermoloop components, other than reservoir are completely filled with working fluid.
ii. No non-condensable gases and air leakage present in the system.
iii. All the device components are at the same level therefore effect of gravity on the thermoloop operation is ignored.
iv. The condition in the evaporator is always saturation condition.
v. The properties are considered constant for the each time period, after a stable periodic operation is reached.
vi. Condenser outlet is always sub-cooled liquid.
vii. The reservoir is closed, flexible and always maintains atmospheric pressure.
viii. The heat loss from different components of thermoloop is by natural convection.

Nomenclature:

m_E : Mass of Fluid in Evaporator at the initial time (gm)
m_{CR} : Mass of fluid entered into condenser during return time (gm)
m_l : Mass of liquid (gm)
m_v : Mass of vapor (gm)
m_s : Mass of Evaporator body (gm)
$\dot{m}_r$: Mass flow rate during return time (gm/sec)

$\dot{m}_s$: Mass flow rate during slug flow time (gm/sec)

$\dot{m}_v$: Mass flow rate during vapor flow time (gm/sec)

T_b : Boiling point of fluid (^oC)

$T_{E\ in}$: Condenser outlet temperature (^oC)

T_{max} : Maximum evaporator wall temperature (^oC)

T_{Steady} : Steady evaporator wall temperature after fluid return (^oC)

T_{slug} : Temperature of evaporator body during slug flow (^oC)

T_{CR} : Condenser inlet temperature during return time (^oC)

C_P : Specific heat capacity of fluid (kJ/kg K)

S : Specific heat capacity of evaporator material (kJ/kg K)

L_V : Latent heat of vaporization (kJ/kg)

Q : Heat Input (W)

P_E : Evaporator pressure (Pa)

ΔP_{N-R} : Pressure drop in non return valve (Pa)

ΔP_{FL} : Pressure drop in the forward line (Pa)

ΔP_{RL} : Pressure drop in the return line (Pa)

P_{Vnec} : Vacuum pressure necessary to return liquid into the evaporator (Pa)

ΔP_C : Pressure drop in the condenser (Pa)

$P_{v\ th}$: Theoretical vacuum in the condenser (Pa)

$P_{v\ act}$: ctual vacuum pressure inside the condenser (Pa)

P_b : Barometric pressure (Pa)

P_s : Partial pressure of fluid vapor (Pa)

t_c : Cycle time (sec)

t_R : Fluid return time (sec)

t_N : No flow (from evaporator) time (sec)

t_S : Slug flow time (sec)

t_v : Vapor flow time (sec)

ρ_l : Density of liquid (kg/m^3)

ρ_v : Density of vapor (Kg/m^3)

μ_l : Viscosity of liquid (µN.s/m^2)

μ_v : Viscosity of vapor (µN.s/m^2)

$\bar{x}$: Average vapor quality

x_{out} : Output quality of vapor

x_{in} : Input quality of vapor

f_B : Blausius friction factor

3.3 Analytical Study of Thermoloop Operation:

As pointed out by Fantozzi et al [27], a pulsating two-phase thermosyphon device has three main components between which heat and mass are transferred. They are- firstly the evaporator, which receives heat from an external source, then the condenser, which dissipates heat and finally the reservoir, which stores the working fluid and supplies it to the evaporator for completing the heat transfer cycle. The operation time of single cycle of the thermoloop device can be divided into four separate time zones- no flow time- in which liquid in evaporator gets heated from subcooled liquid to saturated liquid, slug flow time- in which saturated liquid stats boiling and slug flow occurs from evaporator to condenser, vapor flow time- in which evaporator outlet is single phase vapor and return time- in which stored liquid from reservoir enters into the evaporator through return line.

Figure 3.1 shows the schematic arrangement of the various thermoloop components for the present study. Thermoloop device starts functioning as soon as the heat flux is supplied to the evaporator and the temperature of the evaporator wall exceeds the boiling point of the liquid. Vapor bubble forms and the pressure inside the evaporator increases in an isochoric process. When pressure inside the evaporator exceeds that pressure drop of non-return valve, condenser and forward line then the liquid in the forward (liquid) line is pushed towards the condenser and the vapor from the evaporator replaces liquid in the forward line and a part of the condenser. The movement of liquid or vapor in the other direction (return line) is prevented as the non-return valve between the evaporator and reservoir remains closed. This vapor starts to cool and is subsequently condensed in the condenser.

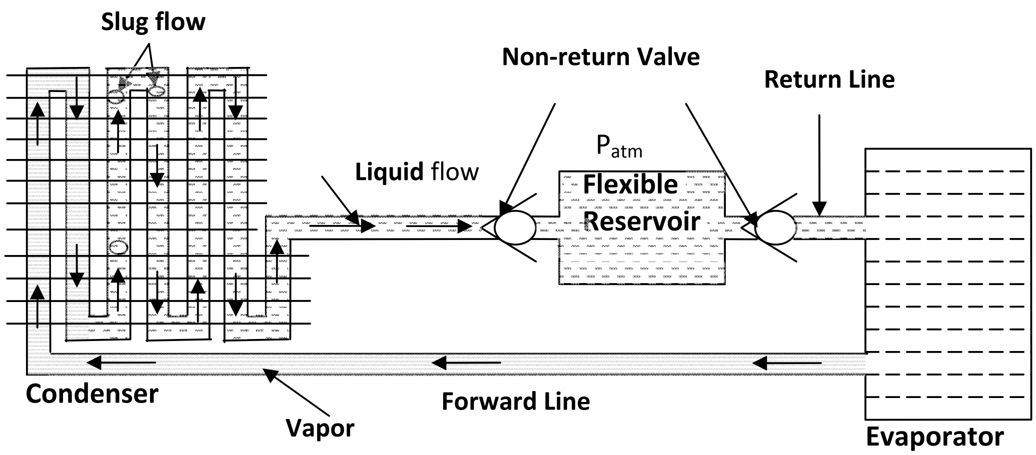


Fig. 3.1: Top view of the experimental setup

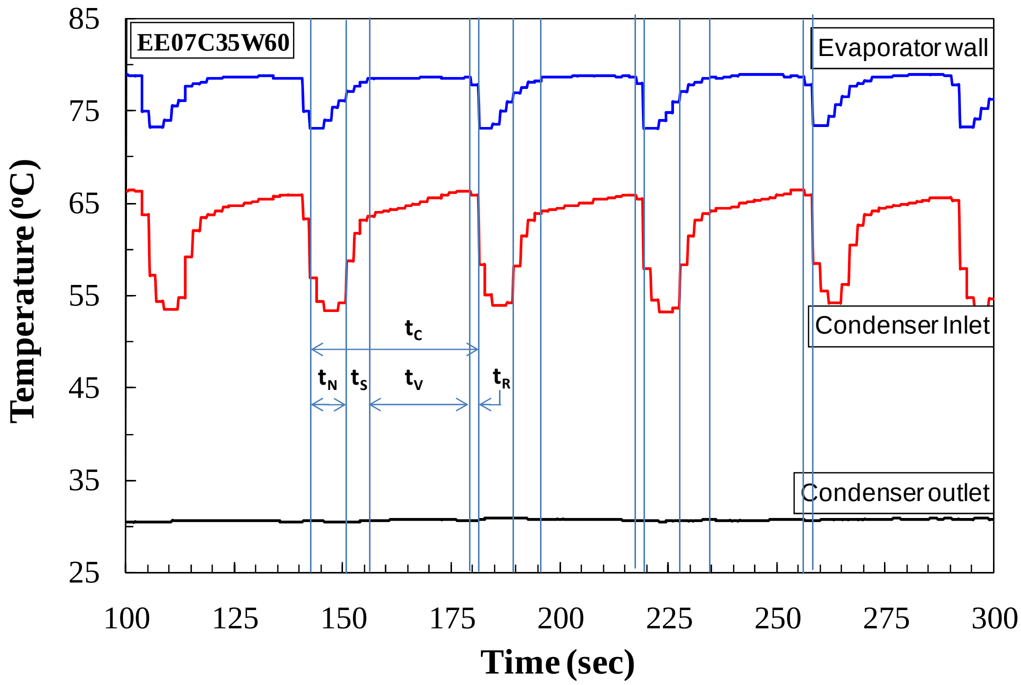


Fig. 3.2 : Heat transport cycles. Working fluid Ethanol, Heat Input 60 W

Figure 3.2 shows how the cycle time is divided into four time zones. Each time zone varies with heat input, fluid and evaporator inner volume.

$$\text{Cycle time, } t_C = t_N + t_S + t_V + t_R \tag{1}$$

t_C, t_N, t_S, t_R *and* t_V are determined from experimental observation.

$$\text{Mass of liquid in Evaporator at initial time of every cycle, } m_E = m_l + m_v \tag{2}$$

Mass flow rate (average):

Mass flow rate during return time (t_C)

$$\dot{m}_r = \frac{m_E + m_{cR}}{t_R} \tag{3}$$

Mass flow rate during slug flow time (t_S)

$$\dot{m}_s = \frac{m_l + \frac{m_l}{\rho_l}\rho_v}{t_S} \tag{4}$$

Mass flow rate during vapor flow time (t_V)

$$\dot{m}_v = \frac{m_v - \frac{m_l}{\rho_l}\rho_v}{t_V} \tag{5}$$

Energy balance of a complete cycle

$$m_E C_p (T_b - T_{E\,in}) + m_{CR} C_p (T_{CR} - T_{E\,in}) + m_v L_v = Q \times t_C \tag{6}$$

Energy balance of evaporator at different time period:

Energy balance during return time:

$$m_E C_p \left(T_{Steady} - T_{E\,in}\right) + m_{CR} C_p (T_{CR} - T_{E\,in}) = Q \times t_R + m_s S\left(T_{max} - T_{Steady}\right) \tag{7}$$

Energy balance during no flow time:

$$m_E C_p \left(T_b - T_{Steady}\right) + m_s S\left(T_{slug} - T_{Steady}\right) = Q \times t_N \tag{8}$$

Energy balance during slug flow time:

$$\left(\frac{m_l}{\rho_l}\right) \rho_v L_v + m_s S\left(T_{max} - T_{slug}\right) = Q \times t_S \tag{9}$$

Energy balance during vapor flow time:

$$\left(m_v - \left(\frac{m_l}{\rho_l}\right) \rho_v\right) L_v = Q \times t_V \tag{10}$$

Mass transfer in Thermoloop mainly occurs due to pressure difference between different components. In Thermoloop device, reservoir always maintains atmospheric pressure where as evaporator and condenser pressure varies with time.

Mass transfer from Evaporator to reservoir through condenser and non-return valve occurs in two stages - Slug flow and vapor flow. During slug and vapor flow pressure inside the evaporator is greater than the accumulated pressure drop inside the condenser, non-return valve and line loss due to flow through the forward line. Pressure rise in the evaporator is a result of rapid vapor bubble formation due to evaporation.

$$P_E > \Delta P_{N-R} + \Delta P_{FL} + \Delta P_C \tag{11}$$

$$\Delta P_C = \left(\frac{2 f_B \left(\frac{\dot{m}}{A}\right)^2 L}{D \rho_l}\right)\left[1 + x\left(\frac{\rho_l}{\rho_v}\right)\right]\left[1 + x\left(\frac{\mu_l}{\mu_v}\right)\right]^{-\frac{1}{4}} + \left(\frac{\dot{m}}{A}\right)^2 . \left(\frac{\rho_l}{\rho_v}\right) . \frac{x_{in} - x_{out}}{\rho_l} \tag{12}$$

Where, L and D are the length and diameter of the condenser pipe respectively.

The pressure drop ΔP_C in the condenser mainly depends on mass flow rate and is a function of geometry and size of the condenser. It is very low if the mass flow rate from the evaporator to the condenser is small. Fantozzi et al. [29] used the following correlation to estimate the pressure drop in the condenser assuming a quality of the vapor x = 1 at the condenser inlet and x = 0 at the outlet.

Since reservoir always maintains atmospheric pressure therefore pressure inside the evaporator and condenser cannot be too higher than the accumulated pressure drop of non-return valve, forward line and condenser pressure drop. This case has been experimentally observed and got the same result.

Specification: EE07C35									
Heat Input (W)	Slug flow time			Vapor Flow time			Return time		
	ΔP_{F-L} (kPa)	ΔP_C (kPa)	ΔP_{N-R} (kPa)	ΔP_{F-L} (kPa)	ΔP_C (kPa)	ΔP_{N-R} (kPa)	ΔP_{N-R} (kPa)	ΔP_{R-L} (kPa)	P_b-P_s (kPa)
50	5.47 E-07	0.22	0.498	1.65 E-03	7.27 E-03	0.498	0.498	3.41 E-08	
60	6.01 E-06	3.45	0.498	2.37 E-03	1.04 E-02	0.498	0.498	5.96 E-07	
70	7.72 E-06	4.60	0.498	3.23 E-03	1.42 E-02	0.498	0.498	4.55 E-07	
80	1.52 E-05	10	0.498	4.21 E-03	1.86 E-02	0.498	0.498	1.11 E-06	
90	1.64 E-05	10.8	0.498	5.33 E-03	2.35 E-02	0.498	0.498	1.11 E-06	
100	1.52 E-05	9.94	0.498	6.58 E-03	2.91 E-02	0.498	0.498	1.36 E-06	

Fluid returns to the evaporator due to the vacuum pressure generated inside the condenser. At the end of vapor transfer stage when all the liquid in the evaporator gets evaporated then no mass flow occurs from the evaporator to the reservoir and the condensation of steady vapor in the condenser causes vacuum pressure inside the condenser and finally the pressure in the evaporator starts to decrease and it reaches the saturation pressure (P_{SAT}) corresponding to the average temperature of the condenser (T_C).

The theoretical vacuum pressure inside the condenser can be given as,

$$P_{v\,th} = P_b - P_s \tag{13}$$

Since no non condensable gases and no air leakage are present in the system,

Therefore, $$P_{v\,th} = P_{v\,act} \tag{14}$$

Vacuum pressure necessary to return the liquid back into the evaporator is

$$P_{V\,nec} \geq \Delta P_{N-R} + \Delta P_{RL} \tag{15}$$

Practically actual vacuum pressure is much greater than necessary vacuum pressure

$$P_{v\,act} \gg P_{v\,nec} \tag{16}$$

This case has also observed experimentally.

CHAPTER 4 RESULTS AND DISCUSSION

During each experiment the wall temperatures of the evaporator, condenser inlet-outlet temperatures, heat input and pressure developed in the system have been recorded. Experiments were conducted for six different types of evaporator, four types of condensers and four types of working fluids as mentioned in the previous chapter. The heat input to the device was varied in an incremental manner to investigate the performance of the device at different thermal loads. The height of the liquid column accumulated in the reservoir is taken as an indicator of the pressure developed in the system.

More than 250 complete sets of data were taken during the course of the present study to comprehend the effect of the different parameters on the performance of the thermoloop device. The observed results for the variation in these parameters are described in details in this chapter.

4.1 The Nature of Heat Transport Cycle

The heat transport cycle of thermoloop device comprises of two repeating states – vapor transfer state and condensate return state. The variation of the evaporator wall temperature, condenser temperatures and accumulated liquid column height in the reservoir during the heat transport cycle is recorded for each step of heat input. Figure 4.1 shows the variation of evaporator wall temperature with time for a heat input of 100 W with water as the working fluid.

From Fig. 4.1, it is clear that the temperature of the evaporator wall increases gradually up to a temperature of 107^0C and boiling initiates when water is used as the

working fluid. It remains almost constant there for a considerable period until all the liquid inside the evaporator evaporates completely. Then the temperature drops to a value of 75^0C when the water from the reservoir flows back into the evaporator- completing the cycle. The same process is repeated in the next cycles. The similar nature in the variation of evaporator wall temperature is observed when ethanol is used as the working fluid. In this case, the temperature of the evaporator wall reaches a maximum value of about 95^0C for the same heat input (Fig. 4.2). In this way, the temperature of the evaporator wall oscillates slightly below and above the boiling point of working fluid. From Figs. 4.1 and 4.2 it can also be observed that the cycle time for ethanol (about 9 minutes) is significantly lower than that for water (about 20 minutes) for the same heat input.

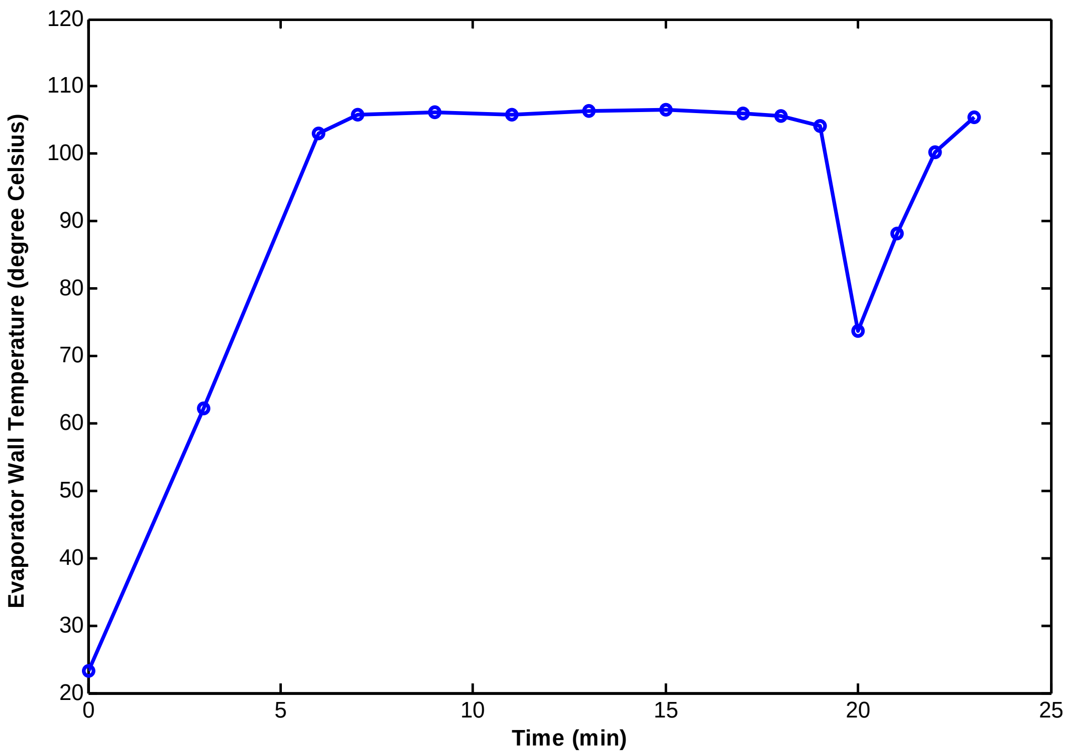


Figure 4.1: Variation of evaporator wall temperature for a heat input of 100 W working with water for E75C50 by using with fan

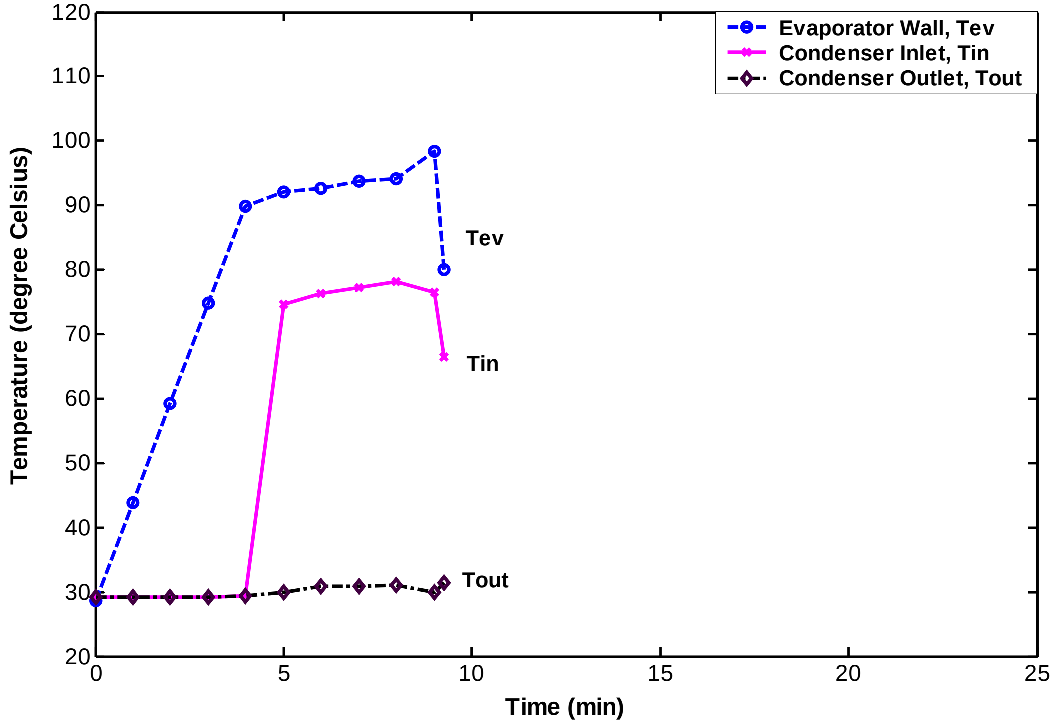


Figure 4.2: Variation of evaporator wall temperature and condenser inlet and outlet temperature for a heat input of 100 W with Ethanol (E75C50, with fan)

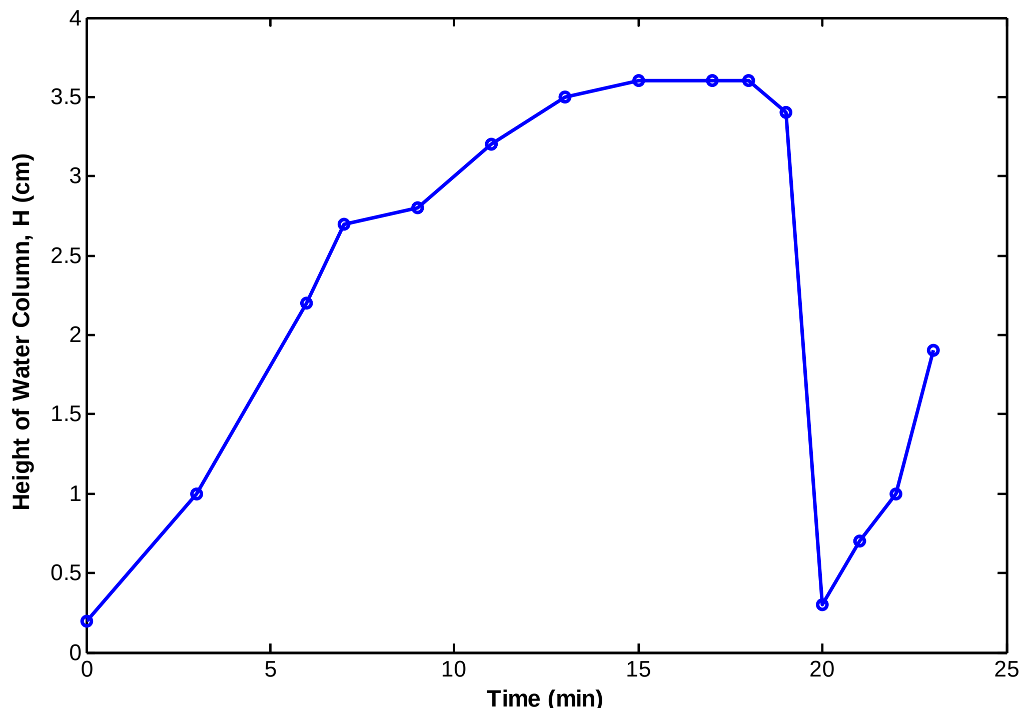


Figure 4.3: Variation of height of water column in the reservoir for a heat input of 100 W working with water for E75C50 using fan

Height of the liquid column accumulated in the reservoir can serve as an indicator of the variation of pressure within the evaporator. Figure 4.3 shows the variation of water column height in the reservoir with time for a heat input of 100 W with water as the working fluid. As soon as the temperature of the evaporator starts to increase with the heat input, liquid in the forward line starts moving forward towards the condenser and then from condenser towards the reservoir by the associated increase in pressure. The liquid is continuously pushed out from the forward line, condenser and liquid line into the reservoir as long as evaporation of the liquid continues inside the evaporator. The water column height in the reservoir then becomes constant for a while when all the liquid inside the evaporator is completely vaporized and no liquid is pushed out of condenser into the reservoir. As soon as the pressure inside the evaporator decreases below the pressure inside the reservoir, liquid flows back quickly into the evaporator through the out valve, leaving the reservoir almost empty and closing the cycle. The temperature of the liquid after coming back to the evaporator starts to increase and the pressure inside the evaporator begins to increase again as vaporization starts. The pressure inside the evaporator continues to pulsate in this periodic manner.

Nature of variation of condenser inlet and outlet temperature with time for the same test is shown in Fig. 4.4. The condenser inlet temperature varies in similar manner as the evaporator wall temperature. It reaches a maximum point after increasing linearly initially, then remains almost steady at this value for some period and then decreases as the cycle completes. <u>But the condenser inlet temperature reaches its maximum value few minutes (about 2 min in this case) after the evaporator wall temperature reaches its maximum steady value.</u> Also the maximum value at which the condenser inlet temperature remains steady is about 90^0C, which is 16^0C lower than the maximum temperature of the evaporator wall for this heat input with water as working fluid. The condenser outlet temperature remains nearly constant throughout the cycle (the maximum variation is less than 2^0C) for this particular test.

The similar scenario is observed in the case of ethanol as the working fluid. The maximum condenser inlet temperature is about 78^0C, which is about 18^0C lower than maximum temperature of the evaporator wall (Fig. 4.2). <u>Also, the condenser inlet temperature reaches its steady pick few minutes after the evaporator wall temperature</u>

reaches its maximum value. The condenser outlet temperature remains almost constant in this case too.

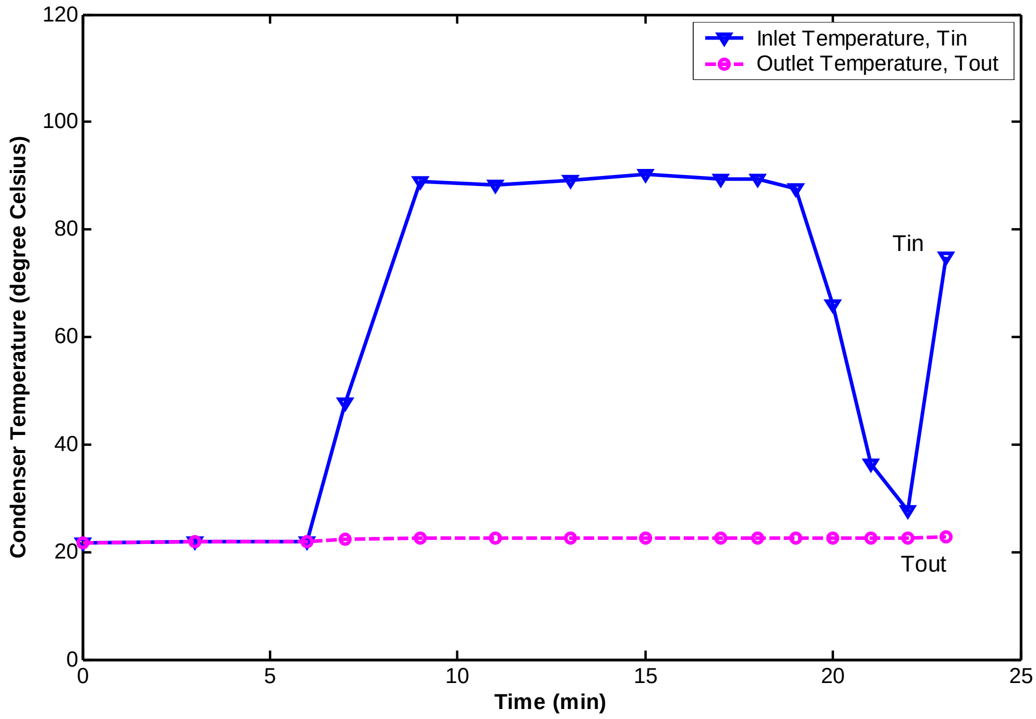


Figure 4.4: Variation of Condenser inlet and outlet temperatures with time for a heat input of 100 W working with water for E75C50 with fan

4.2 Effect of Thermal Load on Various Parameters

After choosing a particular thermoloop device, the first step was to conduct experiments with a view to determining effects of thermal load or heat input on the various heat transfer performance parameters of the thermoloop device. The parameters include the evaporator wall temperature, system pressure, condenser inlet and outlet temperatures, cycle time or cycle frequency, etc. The influence of the variation in thermal load on various parameters of thermoloop is presented below.

4.2.1 Evaporator Wall Temperature

Evaporator wall temperature is the most important operating temperature of the thermoloop as it resembles the temperature of the microchip to be cooled. Figures 4.5 and 4.6 exhibit temperature variations of evaporator wall for an evaporator of 7 cc and a

condenser of 35 cc working with methanol and ethanol respectively. Temperature variations at condenser inlet and out are also shown in these figures. Figure 4.7 shows the effect of thermal load on the maximum temperature of the evaporator wall with water as working fluid. The maximum temperature of the evaporator is plotted for the first 12 heat transport cycles at 4 different heat inputs. The maximum temperature of the evaporator was found to increase with increase in the heat input. The maximum temperature of the evaporator wall was 112^{0}C for a heat input of 250 W, which was about 6^{0}C higher than the same for a heat input of 100 W (106^{0}C). The maximum evaporator wall temperature for the same heat input was found to be almost constant, the variation for different cycle being less than 2^{0}C. It can be noted from the Fig. 4.8 that after the initial unsteadiness, the evaporator wall temperature becomes nearly steady after 4-5 heat transport cycles. This capability of thermoloop device to maintain steady maximum temperature of the evaporator wall is a very important attribute in favor of its applicability for electronics cooling, especially when the electronic component to be cooled is in operation for a long time.

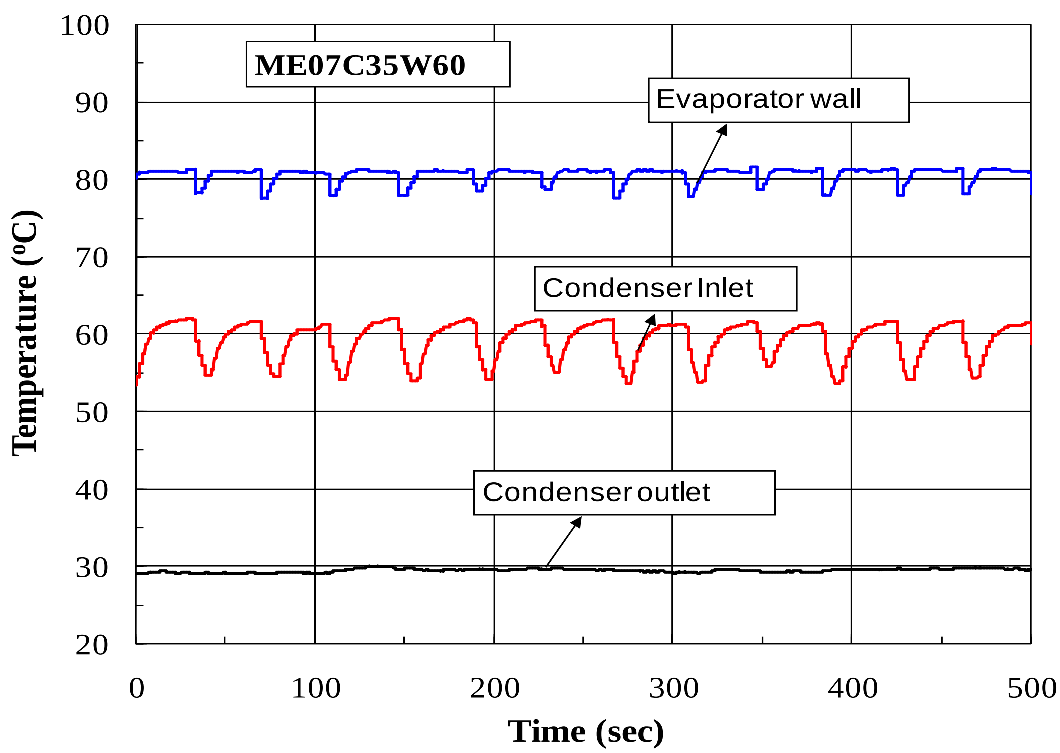


Figure 4.5: Variations of evaporator wall temperature, condenser inlet and outlet temperatures with time for a heat input of 60 W working with methanol for E07C35

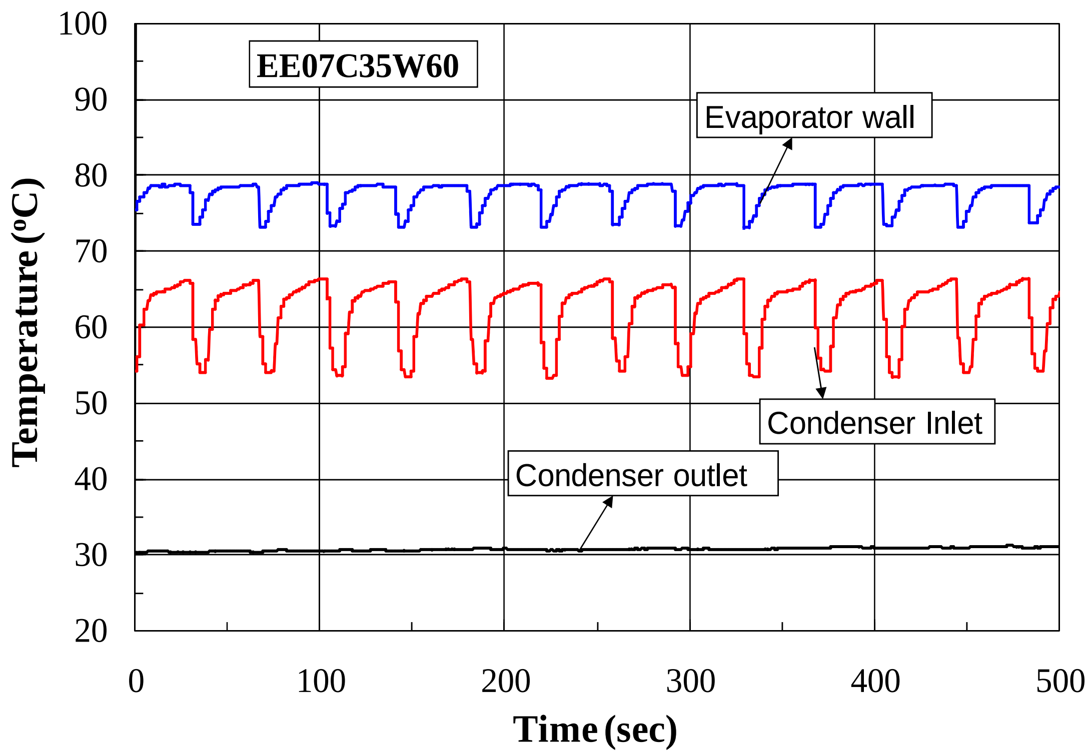


Figure 4.6: Variations of evaporator wall temperature, condenser inlet and outlet temperatures with time for a heat input of 60 W working with ethanol for E07C35

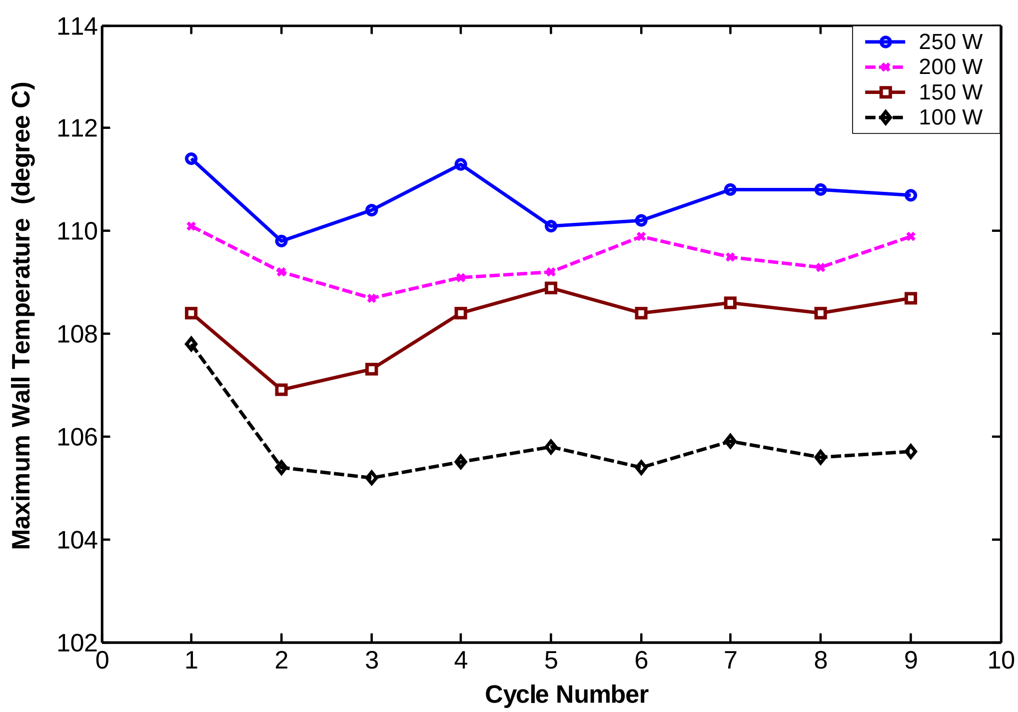


Figure 4.7: Variation of maximum temperature of the evaporator wall for water at 4 different heat loads for E75C50 with fan

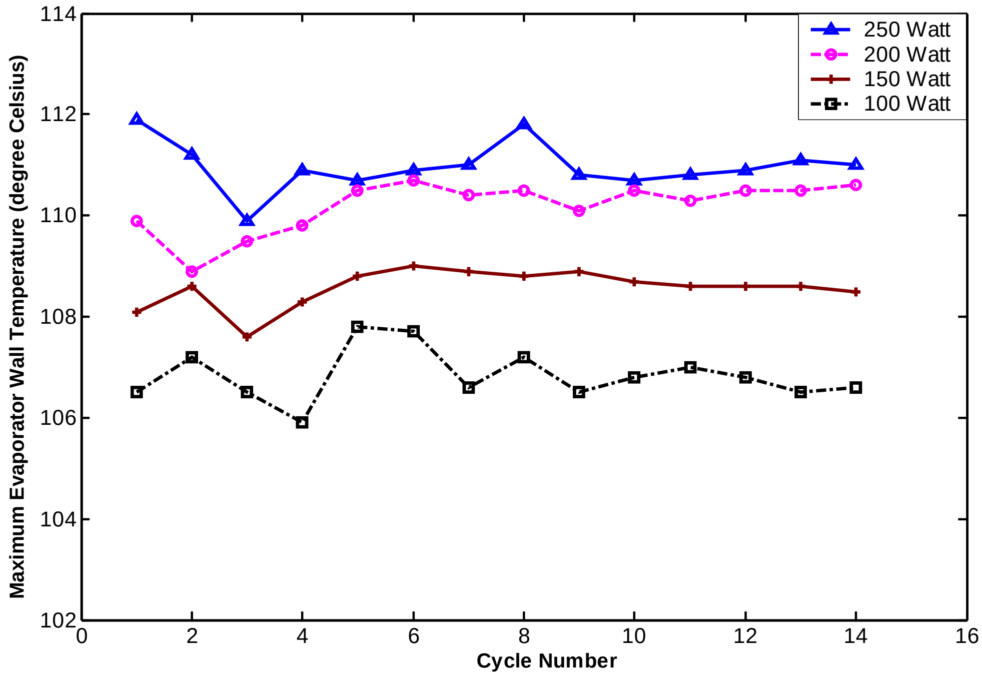


Figure 4.8: Variation of maximum temperature of the evaporator wall for water at 4 different heat input for E75C50 with fan

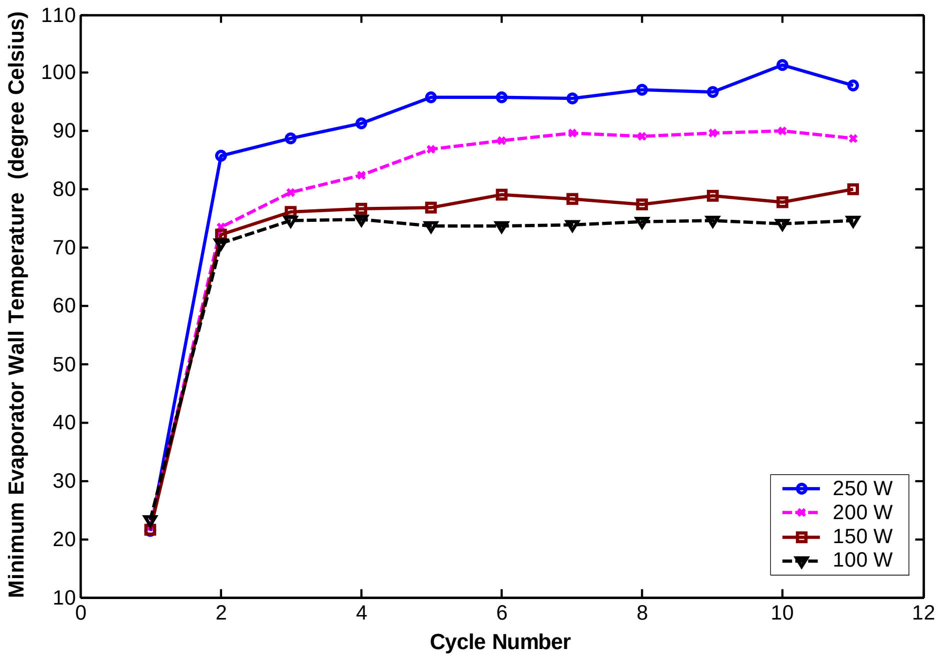


Figure 4.9: Variation of minimum temperature of the evaporator wall for water at 4 different heat loads for E75C50 with fan.

The minimum temperature of the evaporator wall is also a very important parameter because the minimum and the maximum temperature of the evaporator wall indicate the range within which the temperature of the electronic component to be cooled

will The minimum temperature of the evaporator wall is also a very important parameter because the minimum and the maximum temperature oscillate. This variation of minimum wall temperature for a number of heat transport cycles at 4 thermal loads is shown in Fig. 4.9 working with water. The figure shows that the minimum wall temperature increases after the 1st heat transport cycle, and then attains a steady value after 4-5 cycles in a similar manner as maximum wall temperature does. From the Fig.4.9 it is also seen that the minimum temperature of the evaporator wall increases with increasing thermal load. The steady minimum wall temperature was about 93^0C, 90^0C, 78^0C and 74^0C for heat loads of 250 W, 200 W, 150 W and 100 W, respectively.

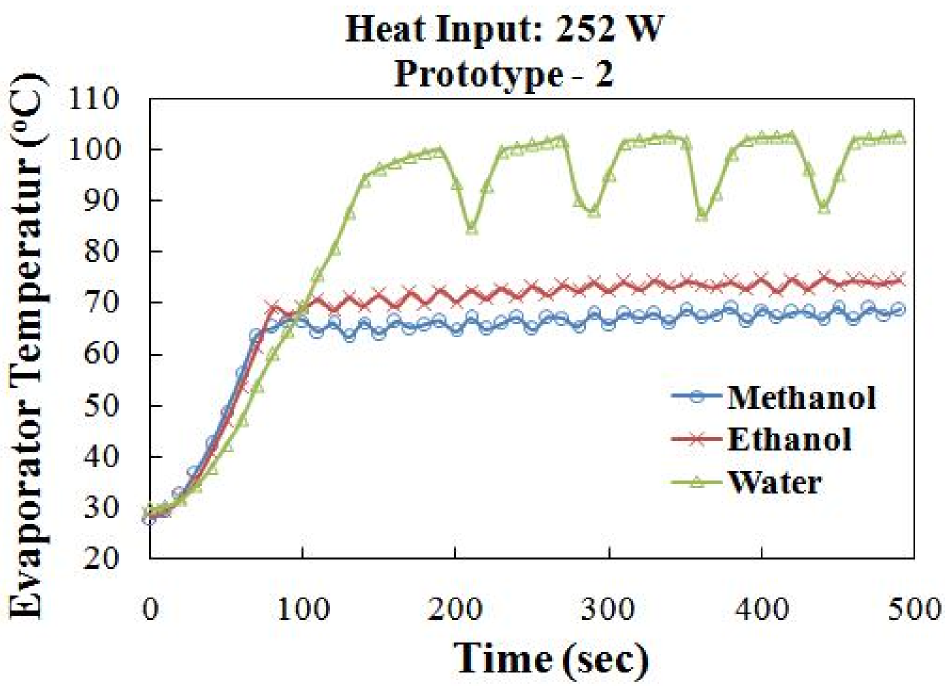


Figure 4.10: Variation of the evaporator wall temperature for different working fluids for E75C50 for a heat input of 250W

Figure 4.10 shows the effects of working fluid on the variation wall temperature for a heat input of 250 W for the E75C50 thermoloop where evaporator volume is 75 cc and condenser volume is 50 cc. It is clear form Fig. 4.10 that water has higher amplitude of temperature fluctuations with lower frequency compared with methanol and ethanol. It is also observed that for lower evaporator volume the evaporator temperature fluctuations are also lower (Fig. 4.11). For lower evaporator volume, condenser with natural convection is enough for necessary heat rejection form the device.

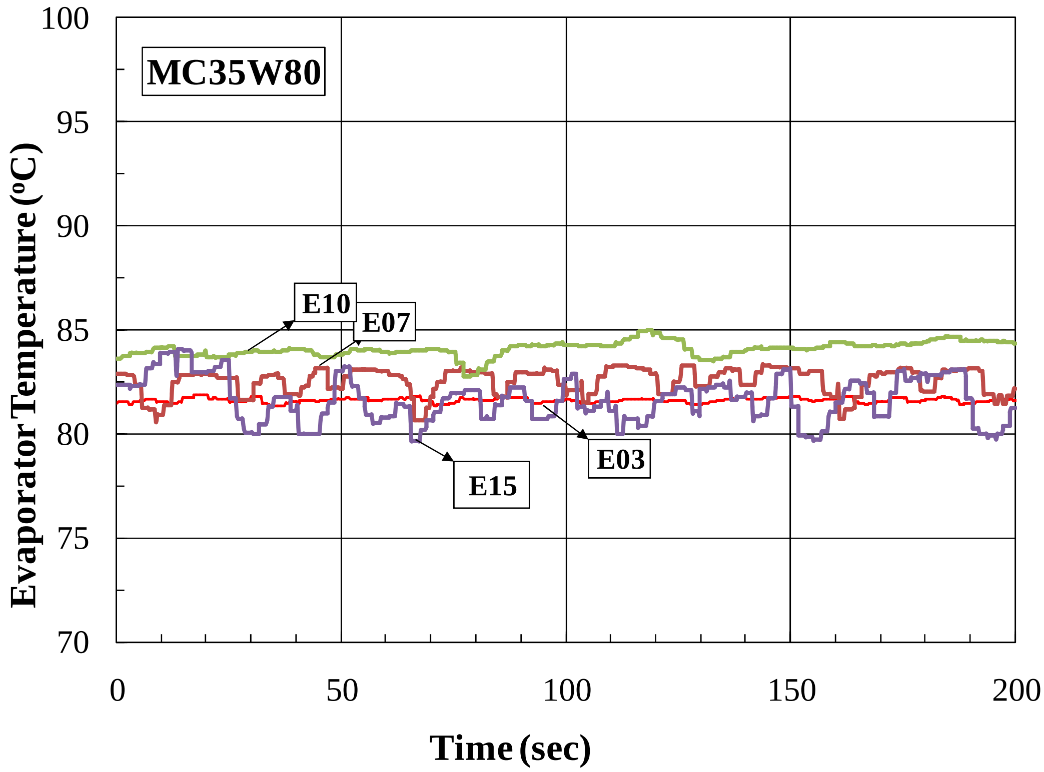


Figure 4.11: Evaporator wall temperature fluctuations for evaporators having different volumes with heat load of 80W working with methanol and a condenser of 35 cc

4.2.2 Condenser Temperature

The variations of condenser inlet and outlet temperatures for different thermal loads are also shown in Fig. 4.12 and Fig. 4.13. The maximum temperature of the condenser inlet is nearly constant for all heat inputs, which is about 90^0C for water. The variation of minimum condenser inlet temperature is plotted in Fig. 4.12 for 4 different heat loads with water as the working fluid and when the condenser was cooled by a fan. From the Fig. 4.13 it is seen that the minimum temperature at the condenser inlet increases with increasing heat input. The minimum condenser inlet temperature attains a steady value after 5-6 cycles. The minimum temperature at the condenser inlet is about 26^0C for a heat load of 100 W while it is about 63^0C for a heat load of 250 W. So the heat dissipation capacity of the condenser reaches a limit as the heat input to the device increases. The variation in the minimum condenser inlet temperature is very similar in nature to the variation of the minimum evaporator wall temperature for different heat inputs (Fig. 4.9).

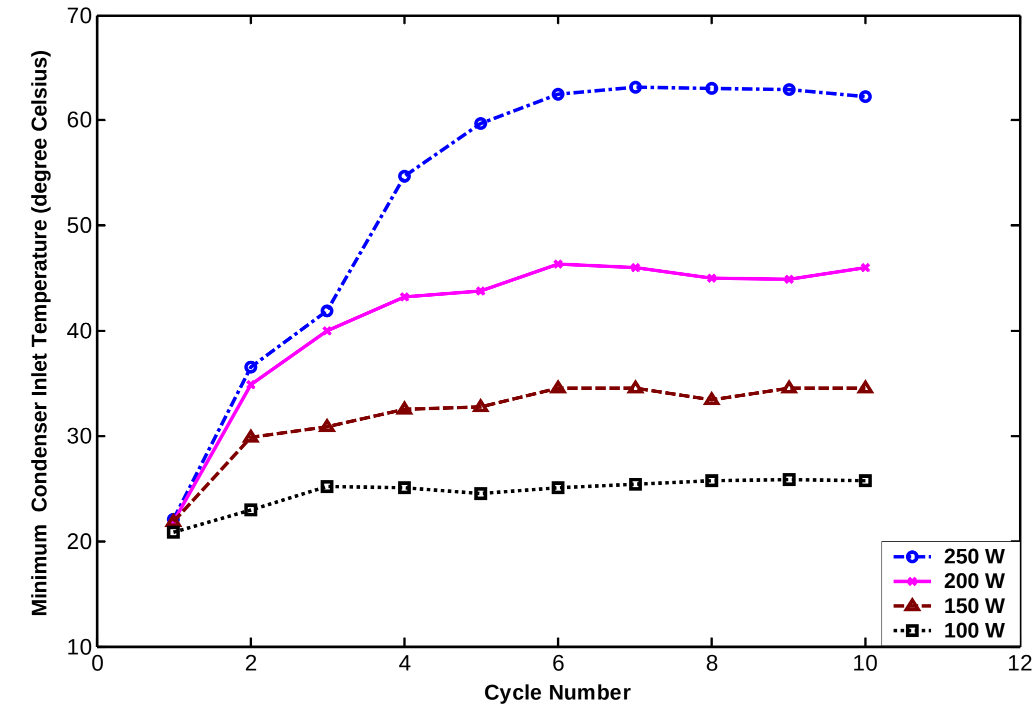


Figure 4.12: Variation of minimum temperature of the condenser inlet at 4 different heat loads with water for E75C50 with fan

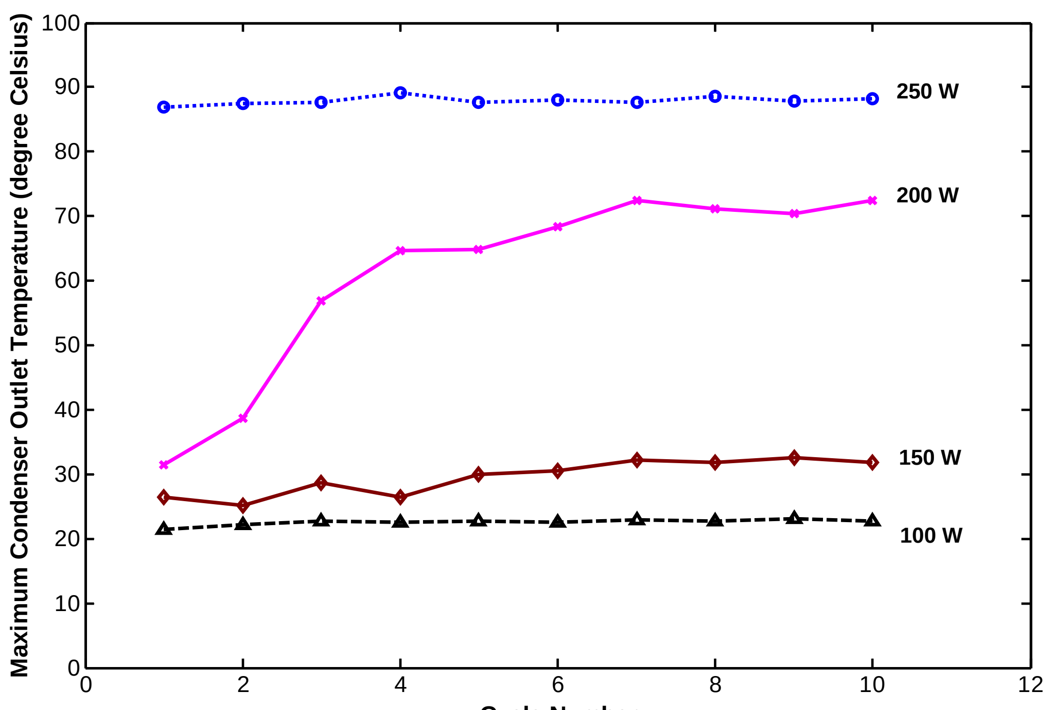


Figure 4.13: Variation of maximum temperature of the condenser outlet at 4 different heat loads with water for E75C50 with fan

The maximum temperature of the condenser outlet is another crucial parameter because it signifies the efficiency of the condenser. The lower the maximum temperature of the condenser outlet, the higher it is able to dissipate heat. Figure 4.13 shows the effect of thermal load on the condenser outlet temperature with water as the working fluid. From the Fig. 4.13 it is seen that the maximum temperature of condenser outlet increases with increasing thermal load. Condenser outlet temperature is almost constant at a value lower than 30^0C for heat inputs of 100 W and 150 W while working with water. But the steady condenser outlet temperature is as high as 65^0C and 87^0C for the heat inputs of 200 W and 250 W respectively.

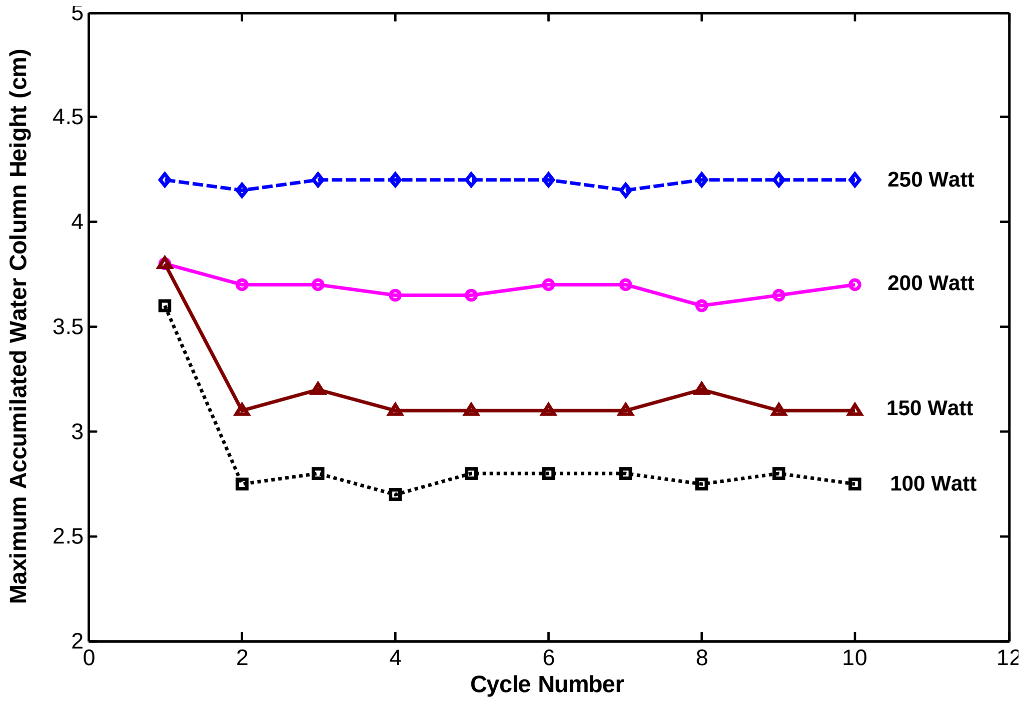


Figure 4.14: Variation of maximum system pressure during operation of the thermoloop device (E75C50) working with water for different heat loads

4.2.3 Height of Liquid Column in the Reservoir

Some of the experiments have been conducted to measure the height of the water column accumulated in the reservoir that is considered as an indicator of the pressure developed within the system. The higher the pressure is developed in the evaporator due to evaporation of the liquid inside it, the greater the amount of liquid accumulated in the reservoir.

Figure 4.3 (as mentioned in section 4.1) clear shows the nature of variation with time of the system pressure during operation of the thermoloop device at a constant heat input. Figure 4.14 shows the effect of variation in thermal load on the maximum height of water column accumulated in the reservoir for different cycles with water as the working fluid. The maximum height of water column increased with an increase in the thermal load. The maximum water column height was also observed to be constant for all the cycles for same heat input. For a thermal load of 250 W, the maximum height is about 4.2 cm while it is about 3.7 cm for 200 W, 3.2 cm for 150 W and about 2.8 cm for a heat input of 100 W. Similar trend of the effect of thermal load on the maximum liquid column height was observed for the other fluids.

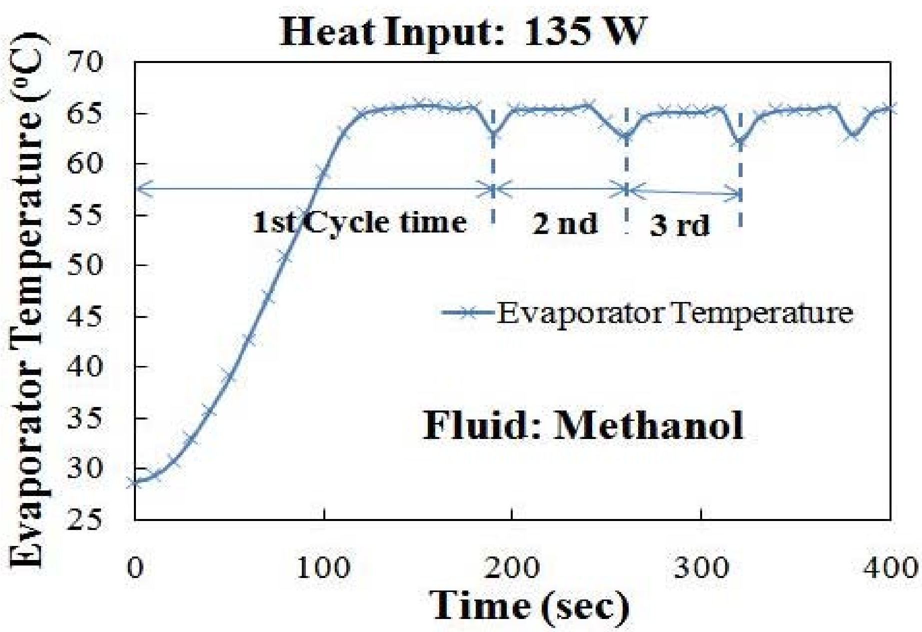


Figure 4.15: Illustration of cycle time

4.2.4 Cycle Time or Frequency

The cycle time is the time necessary to make the liquid back into the evaporator from the reservoir after first being evaporated in the evaporator. For a heat transport cycle, it is taken as the summation of the time required for vapor transport state and condensate return state. It is well illustrated in Fig. 4.15. For a particular heat input, earlier cycle times are much higher than its steady state values as depicted in Figs.4.15-4.16. This is attributed due to fact that the device starts from relatively colder at each level of heat input. The lower the cycle time, the quicker the device completes these two repeating

states. It is greatly influenced by the variation in thermal load. It has a reciprocal relationship with the thermal load as shown in Fig. 4.16 and attains a steady value after 4-5 cycles for all the heat input.

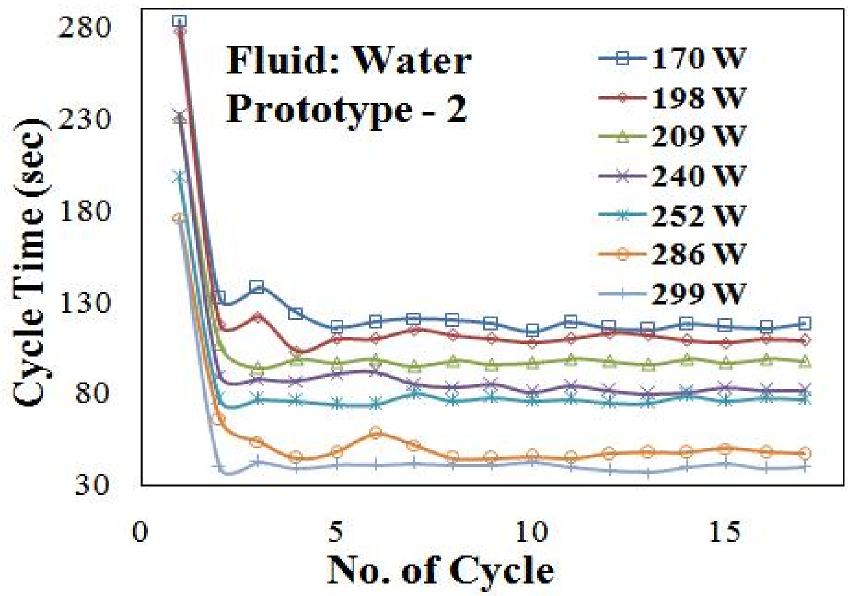


Figure 4.16: Effect of heat load on cycle time during operation for the thermoloop E25C50 working with water

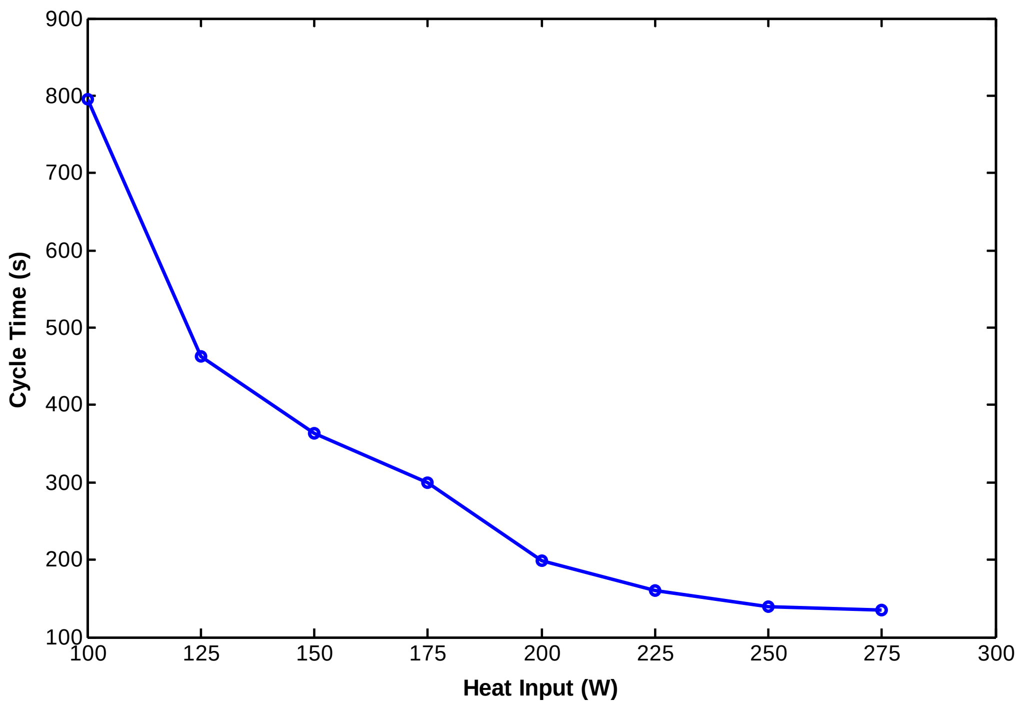


Figure 4.17: Effect of thermal load on the cycle time working with water for E75C50

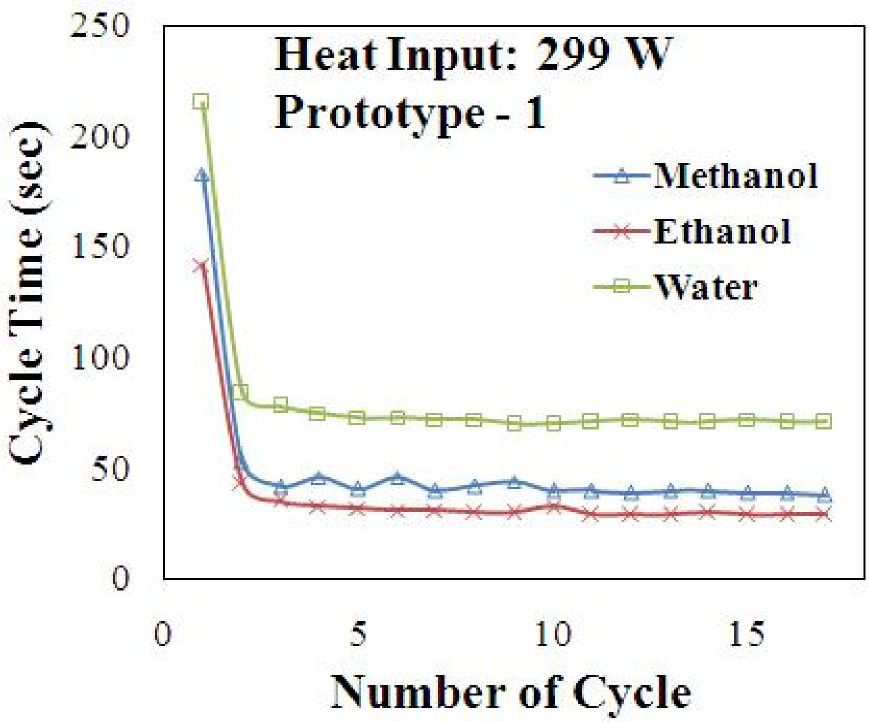


Figure 4.18: Effect of working fluids on cycle time during operation for thermoloop E75C50

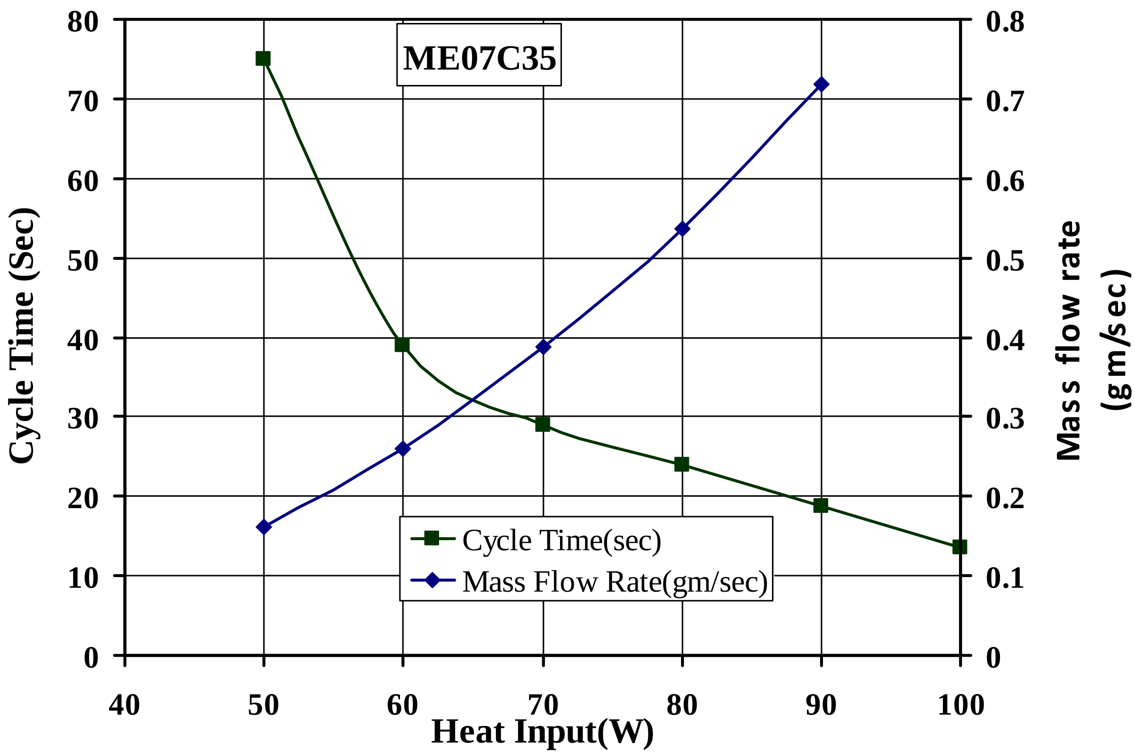


Figure 4.19: Relationship with cycle time and flow rate of the working fluid (Methanol) for E07C35

Figure 4.17 shows the variation of steady cycle time with heat loads for the thermoloop having evaporator of 75cc and condenser of 50 cc working with water. The steady cycle time is about 800 s, 370 s, 200 s and 140 s respectively for heat inputs of 100 W, 150 W, 200 W and 250 W. From this figure, it can be noted that the cycle time decreased rapidly as heat input was increased from 100 W to 150 W, but decreased only moderately as heat input was increased from 150 W to 200 W. After the high heat input of 225 W, cycle time became almost constant. Figure 4.18 shows the variation of steady cycle time with heat loads for the different working fluids for the same thermoloop device. The cycle time has a reciprocal relationship with latent heat of vaporization of the working fluids. Careful investigation may lead to the development of a correlation of the cycle time and/or frequency by considering important properties of the working fluids.

Figure 4.19 demonstrates a relationship with cycle time and flow rate of the working fluid for a device of E07C35 working with methanol. For the higher is the heat input, the lower is the cycle time and the higher is the flow rate of the working fluid.

4.3 Effect of Condenser Convection Conditions

Pal et al. [17] has reported the results of the numerical simulation performed on a compact thermosyphon to reflect on the efficiency of the condenser in natural convection condition and the effect of the system fan on the performance of the condenser. They adopted a system based numerical approach to model the inside of the PC cabinet. For the case with fan, they found that for an average condenser tube wall temperature of 80^0C, the net heat dissipation from the condenser is about 66 W. The same boundary condition was used for the case where the system did not have a fan. It was found that the condenser subjected to natural convection is able to dissipate 60% of the amount of heat dissipated by the condenser subjected to forced convection. So they concluded that the presence of fan creates a big difference in heat dissipation and improves the performance of the condenser a great deal.

In the present study, the effect of convection condition of the condenser on the overall heat transfer performance of thermoloop has been investigated. Also the temperature at the condenser inlet and outlet was measured and its impact on the system

was analyzed. A cooling fan was mounted at a distance of 3 cm from the condenser which generated an average axial air velocity of 4.75 m/s.

4.3.1 Evaporator Wall Temperature

The variation of evaporator wall temperature for both natural convection and forced convection condition of the condenser at a heat input of 100 watt with water as working fluid is shown in Fig. 4.20 for the thermoloop device, E75C50. It can be seen from this figure that the temperature of the evaporator wall did not vary significantly for the variation of cooling condition of the condenser for this heat input. The maximum temperature of the evaporator wall is a fraction higher for the natural convection condition than when a fan is used to cool the condenser. The minimum temperature of the evaporator wall temperature is also higher in the case of natural convection condition. Thus throughout the operation, the temperature of the evaporator wall is slightly higher when the condenser is cooled by natural convection than when a fan is used.

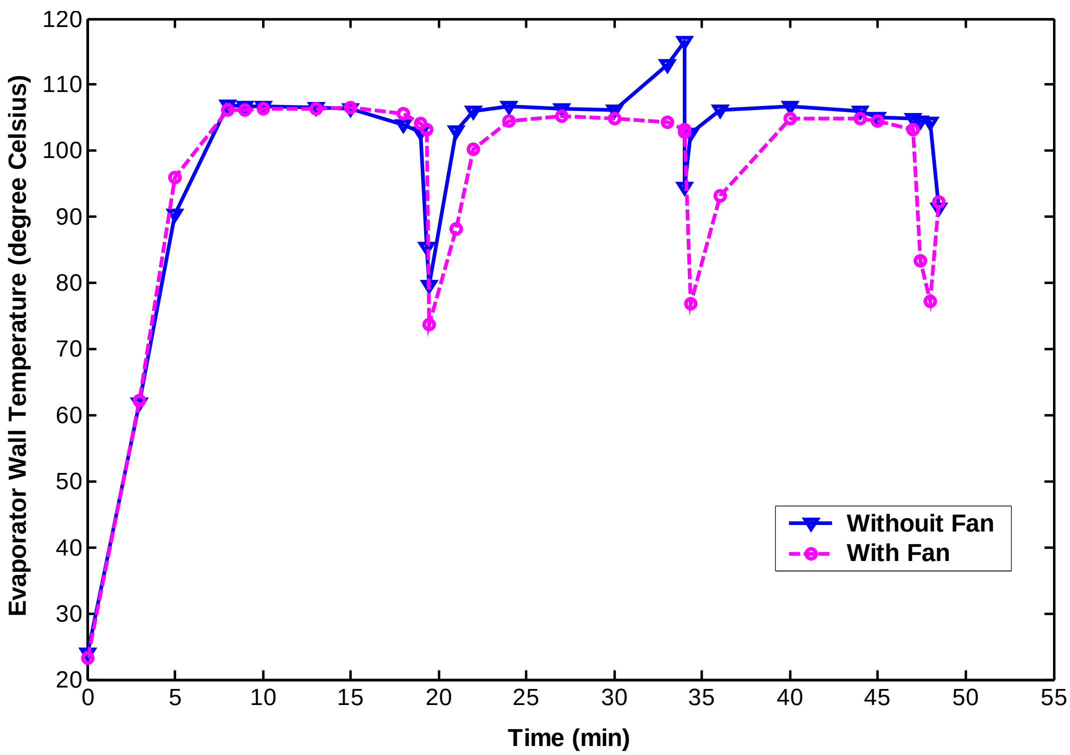


Figure 4.20: Effect of Condenser convection condition on the evaporator wall temperature at a heat load of 100 W of E75C50 working with water

4.3.2 Operating Limit of Thermoloop for Natural Convection Condition

At low heat input such as 75 W, performance of thermoloop was found to be relatively independent of the convection condition of the condenser. But higher heat input has resulted in comparatively poor performance of thermoloop under natural convection of the condenser. Figure 4.20 has shown that the temperature of the evaporator wall was greater in the case of natural cooling of condenser than when a fan was used. In order to find the heat transfer limit of the device, it was tested at a higher heat input.

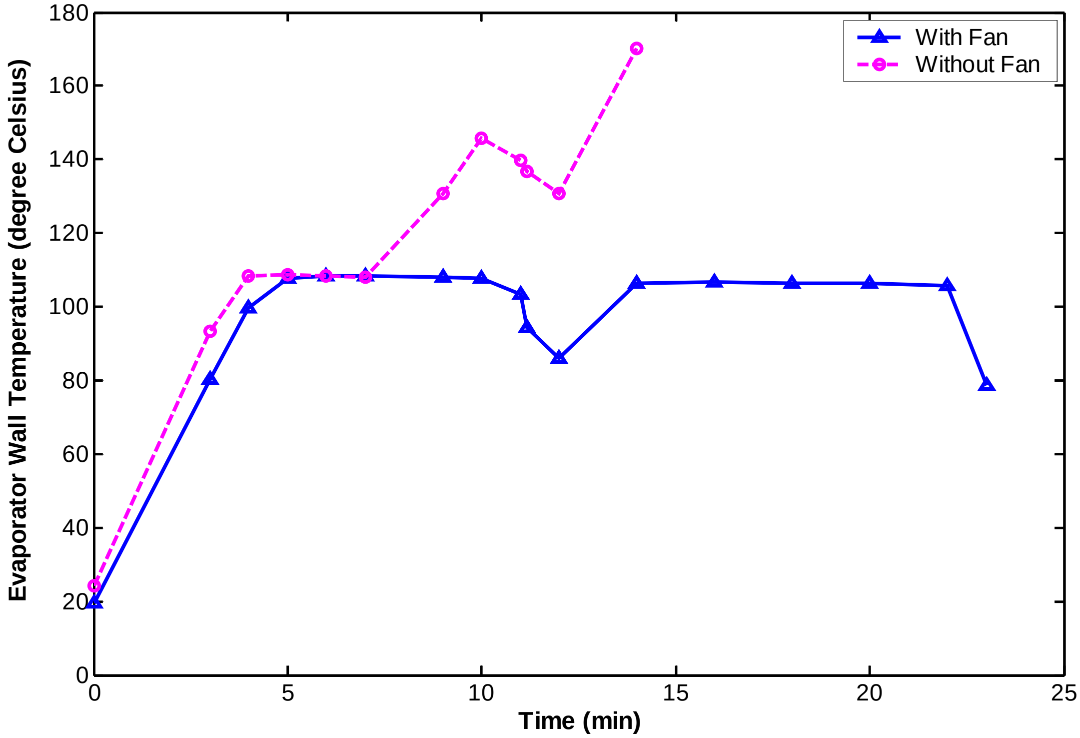


Figure 4.21: Comparison of evaporator wall temperature at different condenser convection condition at heat load of 150 W with water for E75C50

Performance test of the device at a higher load of 150 W with water as working fluid has resulted in the complete failure of the device when the condenser was cooled by natural convection, as shown in Fig. 4.21. The process had to be stopped as the dry out of the evaporator occurred and the temperature of the evaporator wall reached as high as 170^0C. The temperature at the condenser inlet and outlet was nearly equal (both at 97^0C) and the liquid stored in the reservoir was extremely hot. The pressure inside the system was very high as the condenser could not dissipate enough heat from the working fluid.

More tests were performed with a heat input of 135 W with water as working fluid to further investigate the thermal limit of the device without cooling fan. These tests also had to be stopped as the temperature of the evaporator reached a very high value (above 160^0C) within a very short time. The maximum thermal load at which the thermoloop E75C50 can function without fan was found to be 125 watt whereas the same device was operated for a thermal load of 275 W without deterioration of performance when a fan was used.

Condenser cooled by natural convection has higher overall thermal resistance than the forced convection condition due to greater thermal resistance between the outer surface of the condenser and the ambient. It can be expressed mathematically as –

$$(R_{TH})_C = \frac{T_{OUT} - T_\infty}{Q_C} \qquad \textbf{(4.1)}$$

where $(R_{TH})_C$ is the thermal resistance between condenser and ambient, T_{OUT} is the temperature of the outer surface of the condenser, Q_C is the amount of heat dissipated by the condenser and T_∞ is the temperature of the ambient air. From Eq. (4.1) it can be seen that, the higher the temperature of the condenser surface, the higher is the thermal resistance. Temperature of the condenser is very high for natural convection condition, leading to higher thermal resistance and consequently to poor performance.

Table 4.1: Thermal resistance of condenser of E75C50 under forced convection and natural convection conditions for 3 different heat input working with water

Heat input (W)	**Condenser Outlet Temperature T_{OUT} (^{0}C)**		**Condenser Heat Dissipation (W)**						**Thermal Resistance R_{Th} (m^2.K/W)**	
			By Phase Change (Q_{PC})		**By Subcooling (Q_{SC})**		**Total Heat Dissipation, (Q_C)**			
	With Fan	**No Fan**	**With Fan**	**No Fan**	**With Fan**	**No Fan**	**With Fan**	**No Fan**	**With Fan**	**No Fan**
75	23.2	75	57.2	56.7	8.2	2.6	65.4	59.3	0.018	0.89
100	23.4	95	64.3	62.7	9.2	1.8	73.5	64.5	0.019	0.97
125	24.7	96	87.5	72.7	12.5	1.9	100	74.6	0.027	0.99

From Table 4.1, it can be seen that the amount of heat dissipated by the condenser is lower for natural convection condition than when it is cooled by a fan. For the heat input of 125 W, the heat dissipated by the condenser is 100 W for forced convection, while it is only 74.6 W when there was no fan. Heat dissipation due to phase change and amount of subcooling both decrease for natural convection condition, especially at higher heat input. The amount of subcooling without a fan (1.9 W) is only 15 % of the amount of subcooling for forced convection condition (12.5 W) at the same heat input of 125 W, as the condenser outlet temperature is very high (about 96^0C) for natural convection condition. As a result, the thermal resistance of the condenser is about 40- 50 times higher when there was no fan to cool the condenser. Therefore, the thermoloop device reaches its operation limit as it can not dissipate enough heat when it is operated without a fan. Thus the addition of fan had a very significant effect on the heat transfer performance of the thermoloop. Forced convection considerably increased the thermal load limit in which the device can operate successfully. Thus the inclusion of a fan to cool the condenser is a very important requirement for higher heat dissipation rate and greater operating range of the thermoloop.

4.3.3 Condenser Temperature

Variations of condenser inlet and outlet temperatures for the variation in convection condition at a heat input of 100 W are shown in Fig. 4.22 and Fig. 4.23. The condenser inlet temperature varies in a similar periodic manner for both cases. But the maximum temperature at the condenser inlet is about 10^0C higher in all the cycles when there is no fan to cool the condenser (Fig. 4.22). Convection condition at the condenser showed more pronounced effect on the outlet temperature of the condenser. Figure 4.23 shows the variation in this temperature for both natural and forced convection at a heat input of 100 W with water as working fluid. The condenser outlet temperature is varied in a periodic manner similar to that of inlet temperature under natural convection condition and the maximum temperature of the condenser outlet is only 0.5^0C lower than the inlet temperature. So the heat dissipation capacity of the condenser almost reached the limit as both inlet and outlet temperatures of the condenser have reached nearly the same value. But for the case of forced convection, the outlet temperature is very low and it is almost

constant for all the cycles at the room temperature of 25^0C. Thus the forced convection condition offers much higher heat dissipation rate compared to the natural cooling of the condenser.

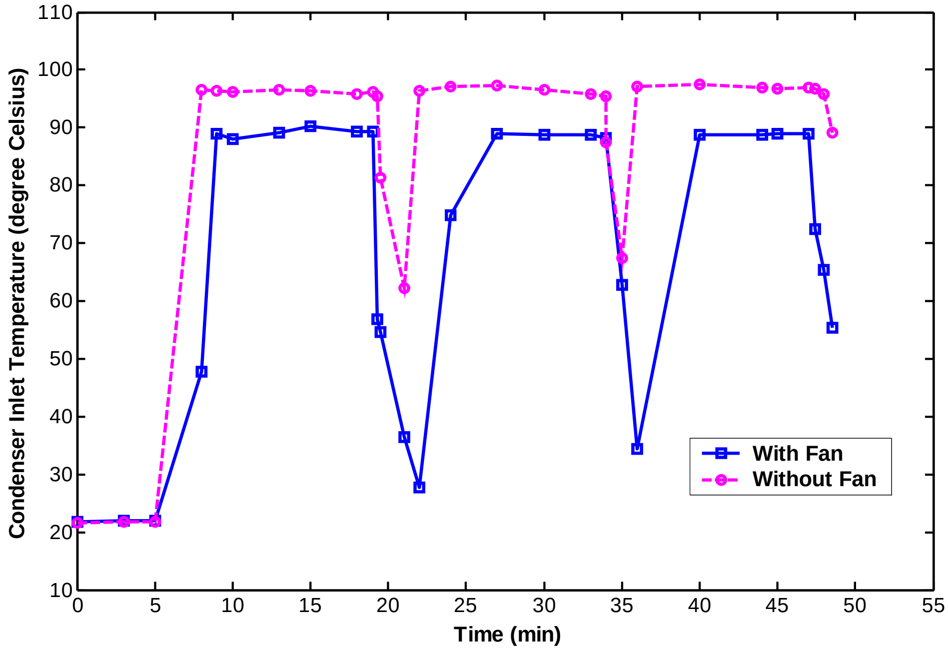


Figure 4.22: Effect of convection condition on condenser inlet temperature for a heat load of 100 W for E75C50

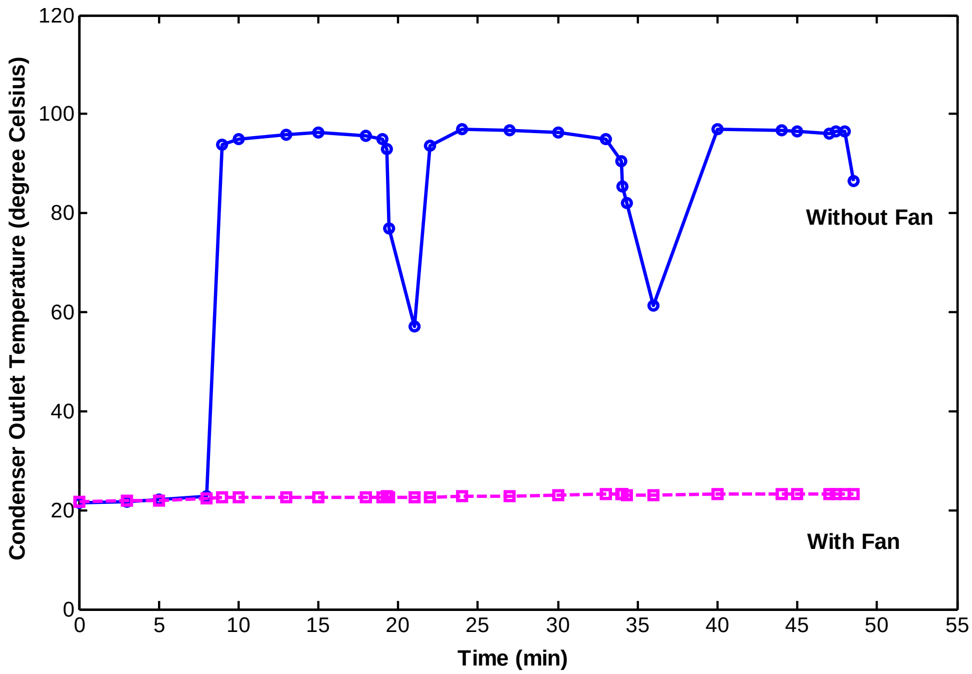


Figure 4.23: Effect of convection condition on condenser outlet temperature for a heat load of 100 W with water for E75C50

4.3.4 Cycle Time

Convection condition of the condenser also exhibited significant effect on the cycle time or frequency. Figure 4.25 shows the influence of cooling condition of condenser on the cycle frequency of the thermoloop device at a heat input of 125 W with water as working fluid. The cycle time decreases more for forced convection than when there was no fan. The steady cycle time for forced convection condition is about 4 minutes less than that of the natural convection condition. But for a lower heat input of 75 W and 100 W, effect of condenser convection condition on the cycle time was less significant. Cycle times for both these conditions were nearly equal for a heat input of 75 W and only a fraction less (less than 1 minute) in the case of a heat input of 100 W. Therefore, it can be concluded from the results that the effect of condenser cooling condition on the cycle time is more prominent for higher heat inputs. The reason is that as the condenser reaches the limit of its heat dissipation rate (inlet and outlet temperature being equal), its efficiency decreases and more time is needed to reduce the pressure of the system. As a result, cycle frequency increases when there is no fan. The effect of condenser convection condition on cycle time for the thermal load of 100 W with water as the working fluid is shown in Fig. 4.26.

4.5 Effect of Leakage in the Device

The most suitable fluid to be used for electronics component cooling by thermoloop is known to be different types of dielectric fluid such as FC-70 or FC-72 etc., as discussed in chapter two. The two working fluids used in the current study were water and ethanol. Water was selected because of its excellent thermophysical properties and wide availability. Ethanol was selected for its low boiling point (necessary for many electronic components' maximum allowable temperature) and its well known suitability for this kind of passive heat transfer device. Both these working fluids adequately meet the demand of this present work of determining the thermal performance of thermoloop. But in the case of these fluids other than the dielectric fluids, perfect sealing of the device has to be maintained as any leakage of the liquid will cause serious damage to the electronic components. In the present work, it was found that apart from the possible damage of the electronic components, the leakage significantly affects on the heat transfer performance of the device.

In the present study, during some tests, leak occurred even after the primary checking to prevent leakage. Leak occurred most frequently in the joints, especially in the connection between evaporator and tubing and also in the insertion point of the two check valves in the return line. The effect of leakage in any part of the device was reflected in the heat transfer nature of the cycle. The major effect of the leakage observed was the notable increase in the cycle time. The liquid coming out of the condenser towards reservoir was observed to move back and forward from condenser in an oscillating manner. The reason is that due to the leakage in the device, the high pressure developed by the evaporation of the liquid could not be maintained within the system. The pressure dropped due to leakage and so the liquid coming out of condenser started to oscillate in and out of the condenser. This pressure pulsation and flow oscillation continued for a long time and the cycle completed with a much higher cycle time. During one or two tests, the oscillation of the liquid back and forward in the return line continued for very long time and the temperature of the evaporator started to shoot off to a very high value, so the process had to be stopped.

Thus the prevention of leakage is extremely important even if the device is operated with dielectric fluid. Leakage not only increases the chance of possible damage of the electronic components and environment, but also can cause the failure of the proper functioning of the device.

4.6 Effect of Flow Controllers

The two flow controllers, placed on either side of the reservoir are very important for the proper functioning of the thermoloop device because they ensure the liquid and vapor flow in the desired directions, which is discussed in chapter two. The first check valve (in valve) maintains that the liquid flow occurs only from condenser to the reservoir and the second check valve (out valve) allows flow from reservoir to the evaporator only. By the proper functioning of these check valves, the return of liquid to the evaporator is made possible. In the present study, the effect of failure of any of these check valves on the performance of thermoloop device was observed during some tests. After continuous use of the same check valves for a number of tests, it was observed that the check valves fail to maintain the flow in the desired direction. This was true especially for the case of the

one way in valve which had to be changed after a few tests as it failed after continuous operation for a number of tests. Back flow from the reservoir through the check valve to the condenser occurred, which is in complete disagreement with the working principle of the thermoloop. The one way out valve also had to be replaced after a certain number of tests, but at a lower frequency than the in valve.

In the case of failure of the in valve, liquid from the reservoir started flowing back into the condenser. But due to the high pressure in the evaporator, this liquid was again pushed back from the condenser towards the reservoir. This pulsation of pressure and oscillation of the liquid continued for a significant period of time. When all the liquid inside the evaporator was vaporized, then pressure started to decrease in the evaporator and the liquid oscillating in the return line between condenser and reservoir flowed back into the condenser. It then started oscillating in and out of the condenser in the forward line, just in front the condenser inlet. This continued for a certain period of time and then the temperature of the evaporator started to rise rapidly and within a few moments, it reached a value as high as 160^0C and the process had to be stopped. Failure in the out valve also caused the failure of the device.

For the proper functioning of the thermoloop device, the flow controllers play an important role by maintaining desired flow directions of liquid and vapor. The plastic flow controllers that were used in the present study failed to work properly after continuous operation and had to be replaced. Therefore, depending on the working fluid and pressure in the system, selection of stronger and proper metallic check valves with the optimum opening pressure is a very important design requirement.

4.7 Thermodynamic Cycle Analysis

From the operation of the thermoloop device, it is understood that as soon as all the liquid in the evaporator is evaporated, condensation of the vapor in the forward line propagates to the evaporator and a vacuum has been generated in the evaporator and a part of the forward line and consequently liquid from the reservoir enters into the evaporator to begin the cycle. This transient behavior of the cycle can be tentatively explained using equilibrium thermodynamics processes and cycles as shown in a T-s diagram given below:

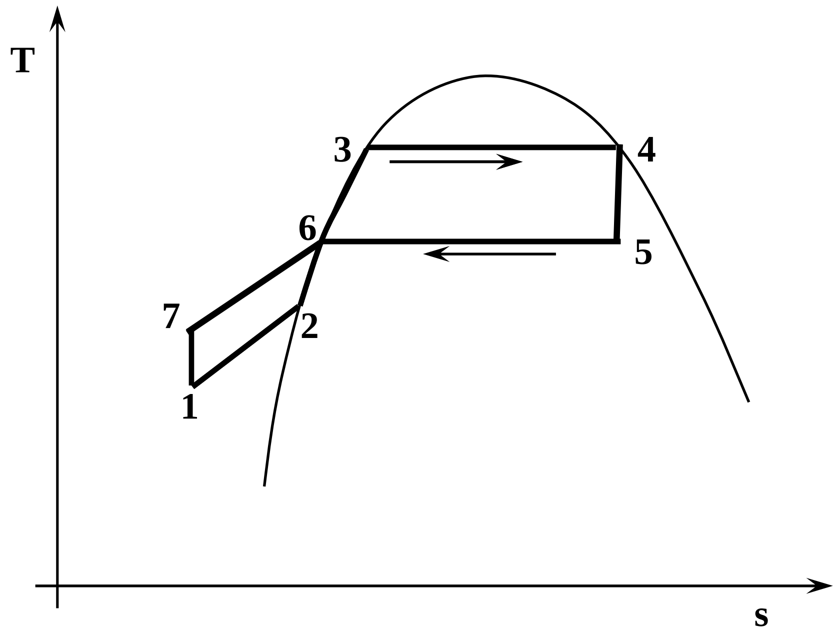


Fig. 4.24: A thermodynamic cycle of the operation of the thermoloop device

Process descriptions:

- 1 - 4: Sensible heating (1-2), pressurization (2-3), vaporization (3-4) in the evaporator
- 4 - 6: Condensation in the condenser
- 6 – 7: Subcooling in the return line and reservoir
- 7 – 1: Liquid from the reservoir enters the evaporator as vacuum pressure generated in the evaporator and a part of the forward line works on it.

The thermodynamic analysis of the cycle 1-2-3-4-5-6-7-1 can be done, if pressure and temperature at all the states are known correctly. The data collected in the present study are not enough to do the analysis. A master study is going on to complete this.

5.1 Introduction

High capacity passive cooling system represents the experimental data at different heat input applied in different evaporator volume with different condenser volume and working fluid.

5.2 High Capacity Passive Cooling System Simulation

This Simulation window represents the following information:

5.2.1 Home Tab

An experiment.gif picture presents the different stages of the Thermoloop device. Thermoloop operations are divided into five stages. First-Initial stage, Second-Liquid transfer stage, Third-Liquid-Vapor transfer stage, Forth-Vapor transfer stage, Five-Liquid return stage. Thermoloop device consist of evaporator, condenser, reservoir, check valve, forward and return line.

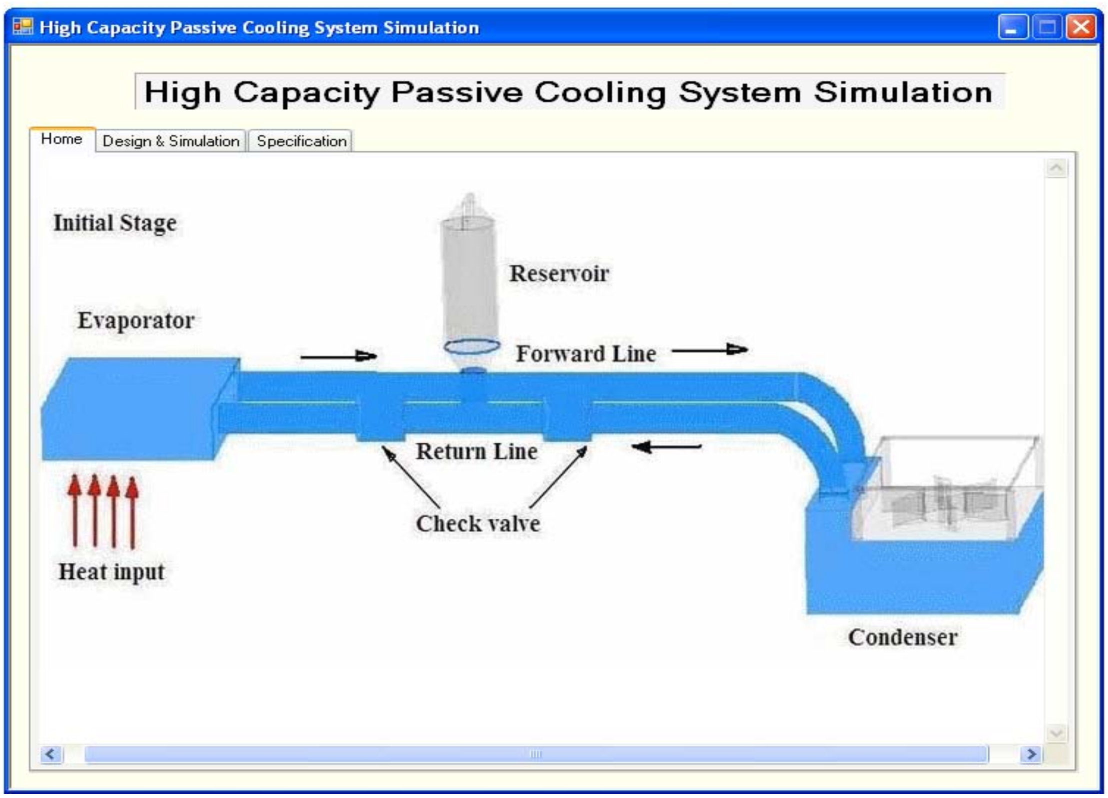


Fig. 5.1: Home window

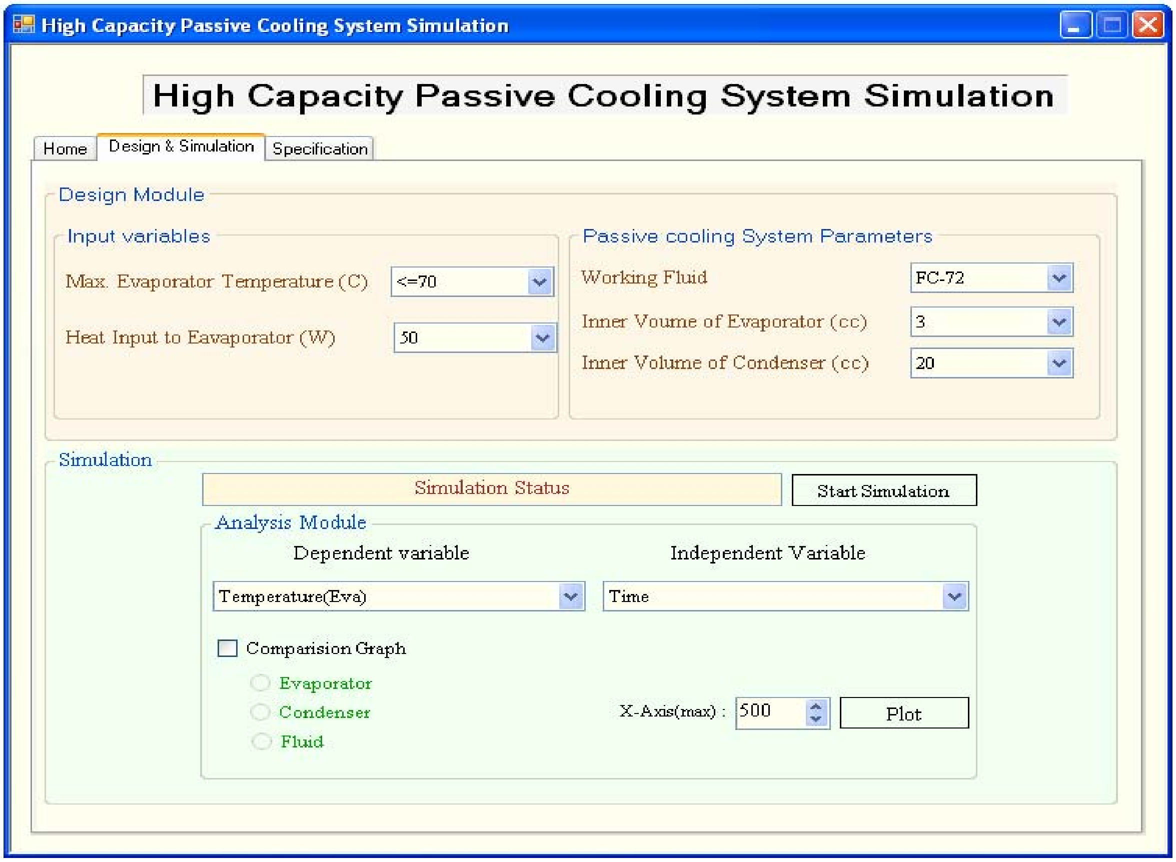


Fig. 5.2: Design and simulation module

5.2.2 Design and Simulation Module

This window represent the experimental data of thermoloop at different volume of evaporator & condenser, heat supply, working fluid and maximum evaporator temperature.

5.2.2.1 Design Module

The design module consists of Input Variables and Passive Cooling System Parameters. Input variable includes:

(a) Maximum Evaporator wall temperature (^{o}C): The range of evaporator wall temperatures is from below 70^{o}C to above 90^{o}C. First -the operator choose the evaporator temperature based on the critical temperature of electronic component.

(b) Heat Input to the evaporator (W): The range of heat applied through the evaporator from 50W to 100W. Secondly-the operator has been chosen the heat input based on the generation of heat from electronic component.

Passive cooling system parameters include three system variables that represent the design of Thermoloop device and maintain the condition of input variable and these are as follows:

(a) Working Fluid: In Design and Simulation, all experiments are conducted by three working fluids. The working fluids are FC-72, Methanol & Ethanol. These fluids are chosen based on their boiling point.

(b) Inner Volume of Evaporator: The inner volume of evaporator represents the amount of working fluid present in evaporator and the size of the evaporator. In Design and Simulation, four different size of evaporator are used. These are 3cc, 7cc, 10cc and 15cc.

(c) Inner Volume of Condenser: The inner volume of condensers represents the amount of working fluid present in condenser and the size of condenser. In Design and Simulation, three different size of condenser are used. These are 20cc, 25cc & 35cc.

5.2.2.2 Simulation Module

At Simulation, Simulation status represents the experimental data of evaporator temperature with time. By pressing **'Start Simulation'**, showing the evaporator temperature (^{o}C) with time(s) collected from database. At Analysis Module represents the graphical relationship of Evaporator Temperature (^{o}C) Vs Time (s) and Evaporator Temperature (^{o}C) Vs Heat input (W) based on the design module. By selecting **'Plot',** graphical presentations of Evaporator Temperature (^{o}C) Vs Time will be shown.

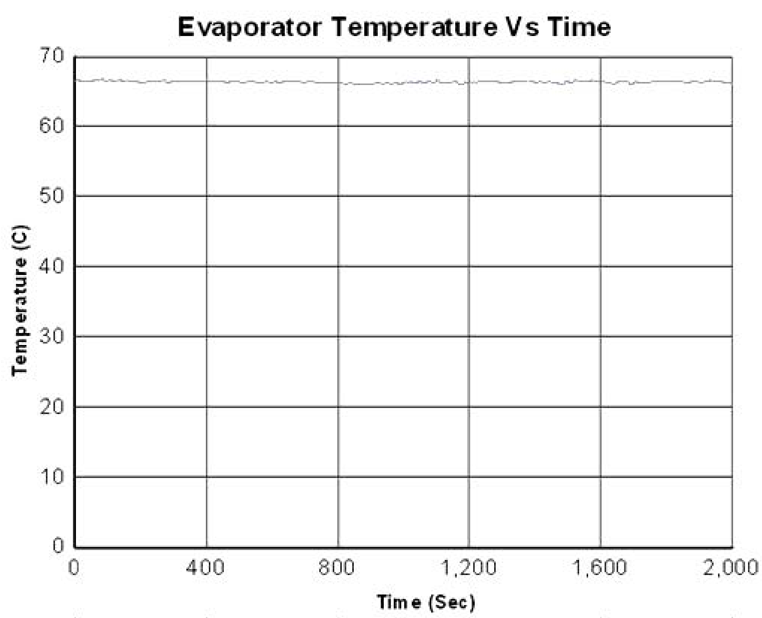


Fig. 5.3: Evaporator temperature fluctuation with time for E03, C20, W50, FC-72

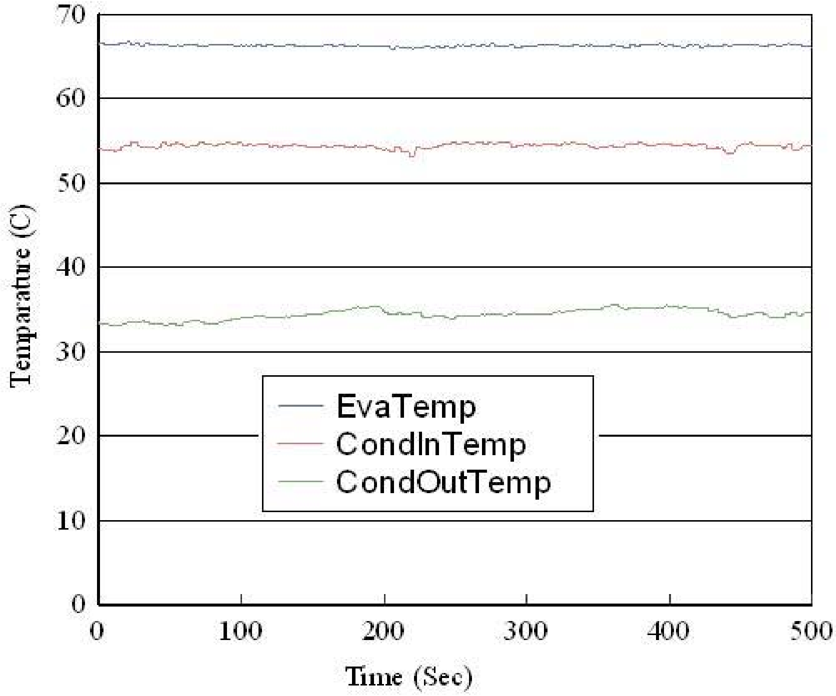


Fig. 5.4: Evaporator Temperature, Condenser inlet and Condenser Outlet Temperature fluctuation with time for E03, C20, W50, FC-72

By selecting Comparison Graph, a graphical comparison of different evaporator, Condenser and Working fluid will be shown.

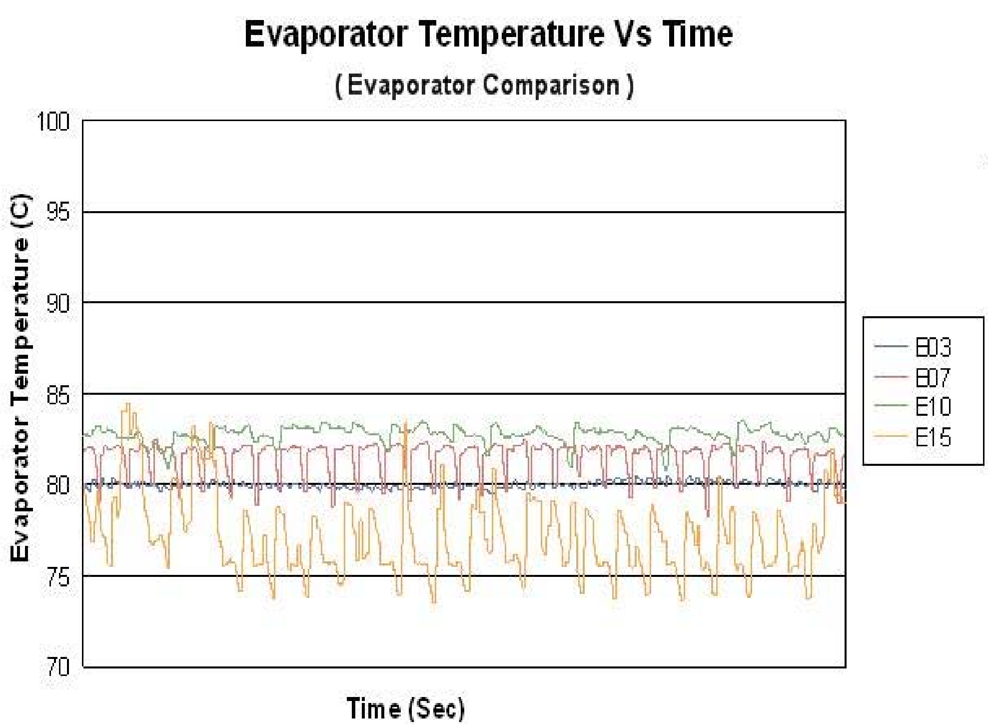
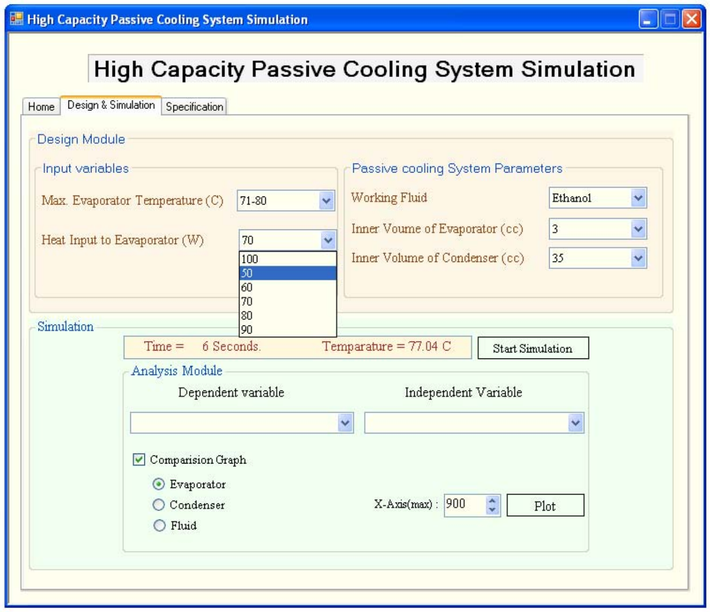


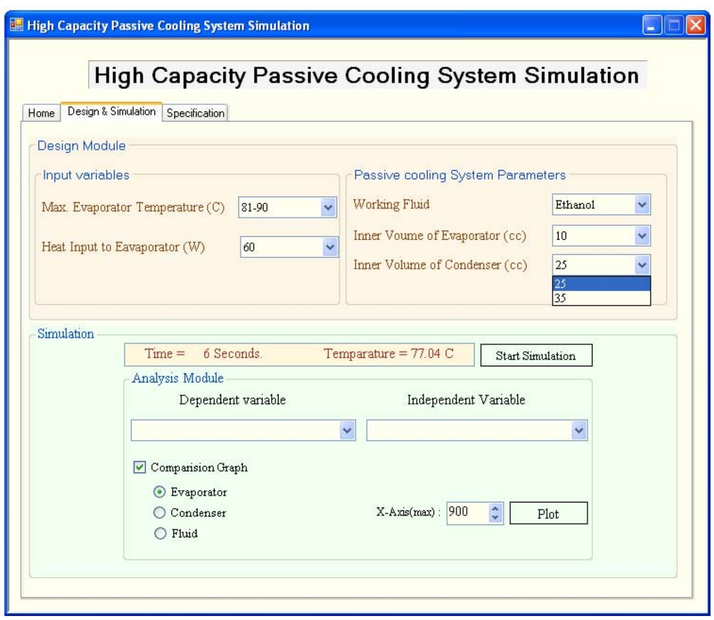
Fig. 5.5: Evaporator Temperature Vs time at different Evaporator volume

5.3 Operating Manual

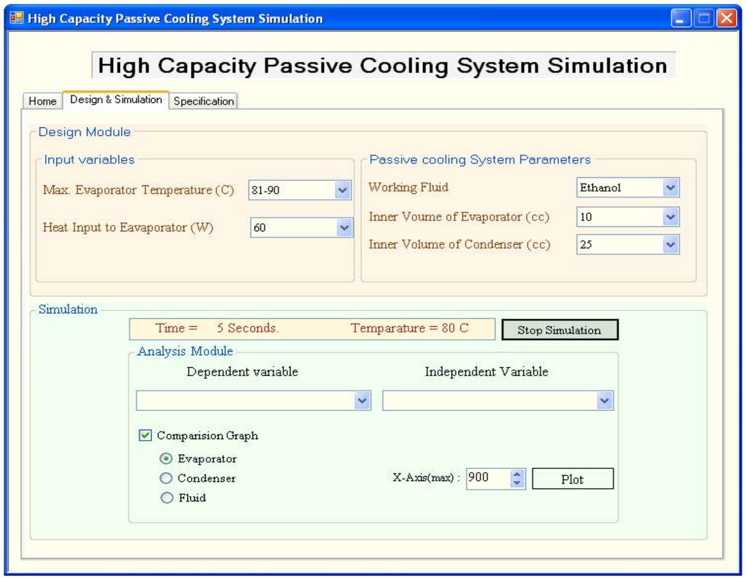
Step-1: Select the Input variables of maximum Evaporator Temperature and Heat input to Evaporator.



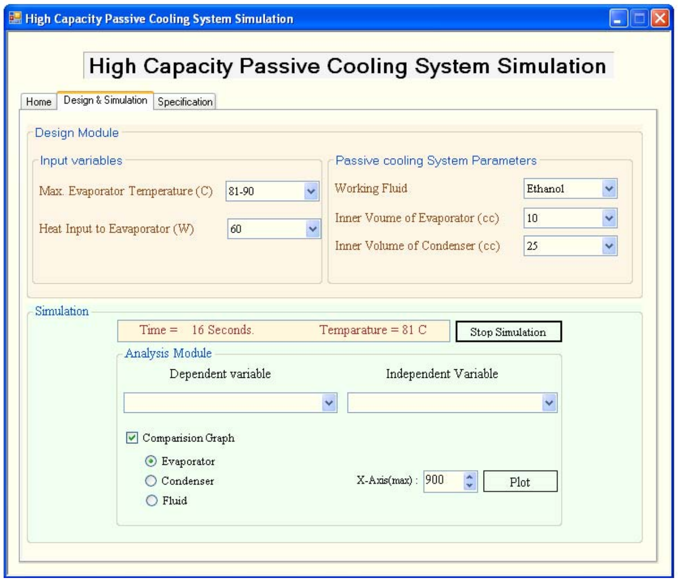
Step-2: Select the Passive Cooling system Parameters of Working fluid, Inner volume of Evaporator, Inner Volume of Condenser.



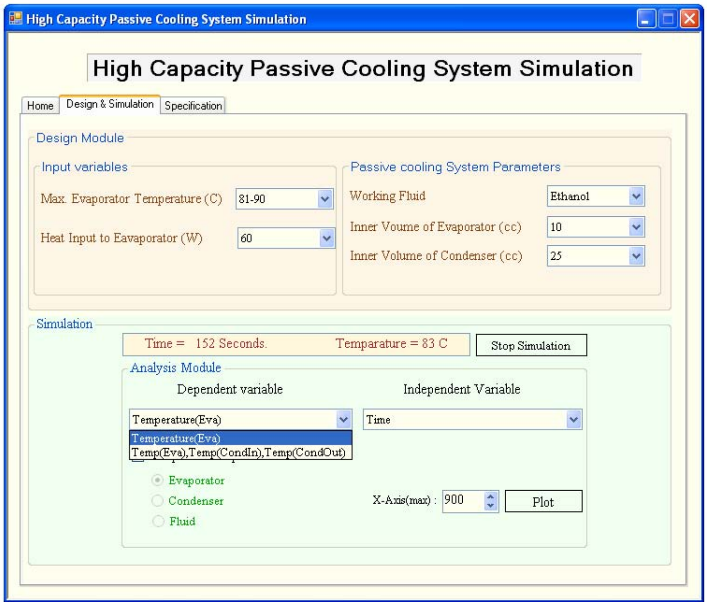
Step-3: Click **'Start Simulation'** as running the simulation.



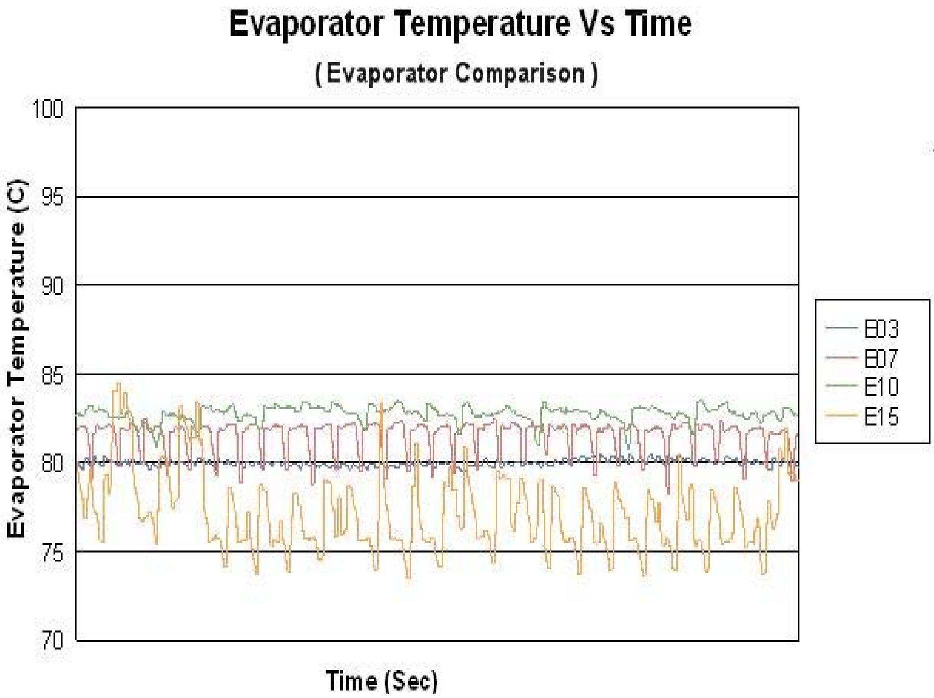
Step-4: Click **'Stop Simulation'** before analyzing module.



Step: 5 From Analyze module, select Dependent Variable and Independent Variable. By pressing **'Plot'**, a graph will be shown which represent Evaporator Temperature Vs Time or Temperature Vs Time of Evaporator, Condenser Inlet, Condenser Outlet.



Step-6: For Comparison among Evaporators or Condensers or Fluids, click any one of them and then click **'Plot'**, a graph will be shown that represent the comparison among the selection component.



CHAPTER 6 CONCLUSIONS

High capacity passive cooling system studied in this project is a thermoloop device that utilizes the thermoloop heat transfer concept (THTC). This device is an assemblage of Evaporator, Condenser, Non-return valves and Reservoir charged with a liquid for removing heat from any source upon which the evaporator is attached.

Extensive experiments are performed to with a view to understanding physics of the device by varying Evaporator volume, Condenser volume and Working fluids. Evaporator wall temperature fluctuations, fluctuation frequency (reciprocal of cycle time), condenser inlet and outlet temperatures are recorded during each experiment. It is found that working fluids having higher latent heat of vaporization have higher amplitude of temperature fluctuations with lower frequency. For example, Water has higher amplitude of temperature fluctuations with lower frequency compared with Methanol and Ethanol. Again, the higher the evaporator volume, the higher is the amplitude of fluctuation and the lower is the frequency. The temperature fluctuation with higher amplitude and lower frequency is always undesirable for electronic system cooling. Therefore, evaporator having smaller inner volume is better option for this device as far as temperature fluctuations are concerned.

Separate mathematical models for both evaporator and condenser are developed to evaluate heat transfer performance on the basis of thermodynamics and heat transfer. Estimating the flow rates of liquid and vapor, and the variation of system pressure correctly during the operation of the device are important challenges for the performance evaluation. Suitable instrumentation may help estimate these correctly in future.

A software module has been developed to choose a suitable device for removing a specified heat at a particular temperature. This module can simulate and analyze data for optimum performance of the device on the basis of experimental data taken so far.

This study ends up with conclusions as listed below:

a) The thermoloop device investigated in this project has the potential to be a better alternative for cooling electronic devices.

b) Based on extensive experimentation, a software module has been developed to analyze the performance of the device.

c) More extensive studies are necessary to mimic the operation of the device by a complete mathematical model.

PROJECT SUMMARY

1. Title and References

Modeling of High Capacity Passive Cooling System

Ref: AOARD-08-4029 Award No: FA4869-08-1-4029

2. Awarded by: The US Air Force Research Laboratory (AFRL)
Asian Office of Aerospace R & D (AOARD)
7-23-17 Roppongi, Minato-ku, Tokyo 160-0032, JAPAN

3. Program manager: Dr. Rengasmy Ponnappan

4. Principal Investigator: Dr. Md. Ashrfaul Islam, Professor, Department of MechE
BUET, Dhaka-1000, BANGLADESH

5. Period: From **06 March 2008** to **06 March 2009**

6. Research Background

High capacity passive cooling system studied in this project is a thermoloop device that utilizes the thermoloop heat transfer concept (THTC). This device is an assemblage of Evaporator, Condenser, Non-return valves and Reservoir charged with a liquid for removing heat from any source upon which the evaporator is attached. This project is to have two major modules: **Design Module–**Device components, materials and working fluids will be chosen on the basis of experimental results and better heat transfer performance and **Analysis Module–**Performance of the device at different thermal loads and ambient conditions to address the relationship among pressure limit, heat flux limit, operating temperature, fluctuation frequency, and fluid physical properties.

7. Research Methodology

Extensive experiments have been conducted and using experimental results a software module has been developed for to analyze the performance of the device for different operating conditions.

8. Research implementation and results

During each experiment the wall temperatures of the evaporator, condenser inlet-outlet temperatures, heat input and pressure developed in the system have been recorded. Experiments were conducted for six different types of evaporator, four types of condensers and four types of working fluids as mentioned in the previous chapter. The heat input to the device was varied in an incremental manner to investigate the performance of the device at different thermal loads. The height of the liquid column accumulated in the reservoir is taken as an indicator of the pressure developed in the system.

More than 250 complete sets of data were taken during the course of the present study to comprehend the effect of the different parameters on the performance of the thermoloop device. The observed results for the variation in these parameters are described in the final report. Evaporator wall temperature is the most important operating temperature of the thermoloop as it resembles the temperature of the microchip to be cooled. Figures 1 and 2 exhibit temperature variations of different temperatures with time for an evaporator of 7 cc and a condenser of 35 cc working with methanol and ethanol respectively.

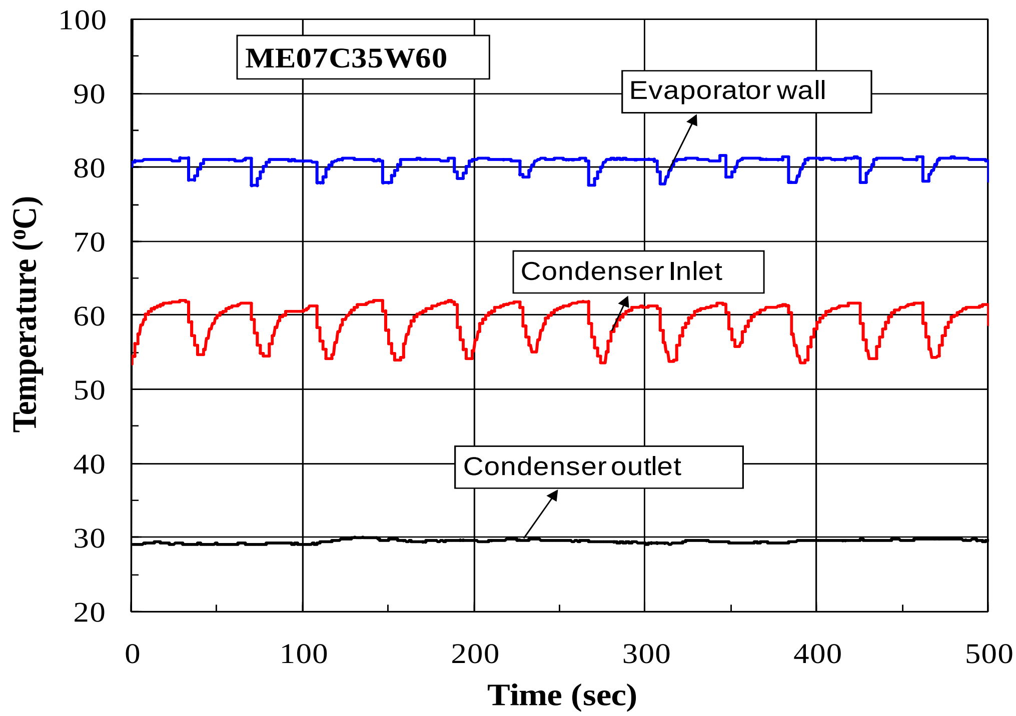


Figure 1: Variations of different temperatures with time for a heat input of 60 W working with methanol for E07C35

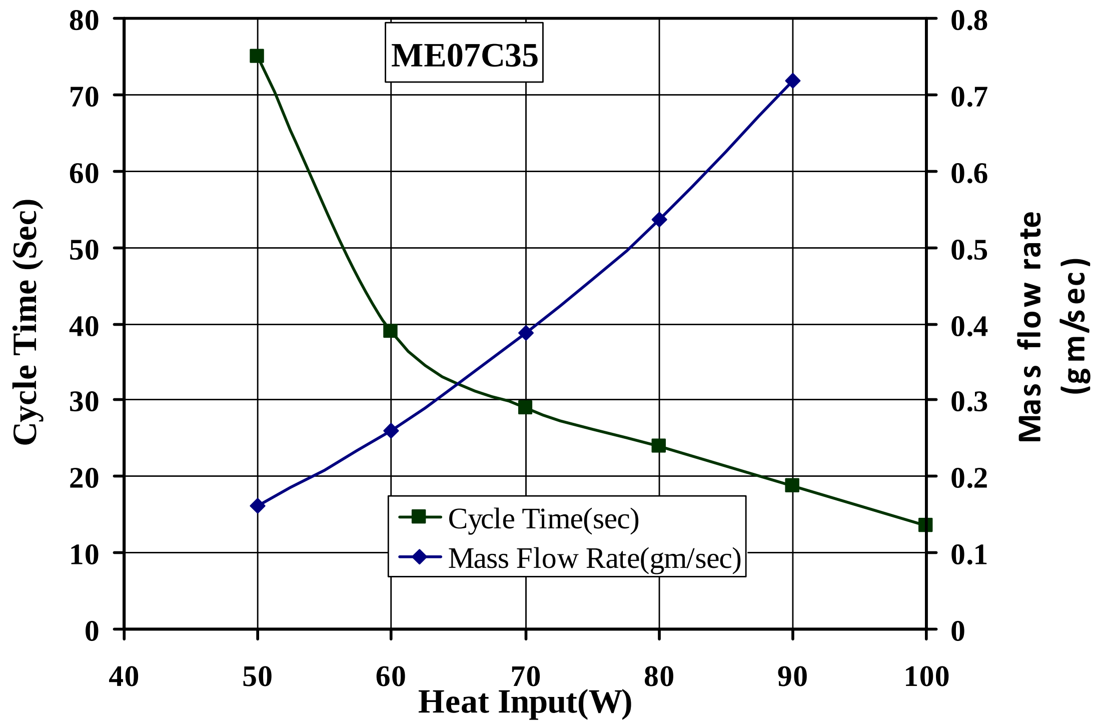


Figure 2: Relationship between cycle time and flow rate of the working fluid (Methanol) for E07C35

The cycle time is the time necessary to make the liquid back into the evaporator from the reservoir after first being evaporated in the evaporator. For a heat transport cycle, it is taken as the summation of the time required for vapor transport state and condensate return state. Figure 2 demonstrates a relationship with cycle time and flow rate of the working fluid for a device of E07C35 working with methanol. For the higher is the heat input, the lower is the cycle time and the higher is the flow rate of the working fluid.

9. Conclusions

- The thermoloop device investigated in this project has the potential to be a better alternative for cooling electronic devices.
- Based on extensive experimentation, a software module has been developed to analyze the performance of the device.
- More extensive studies are necessary to mimic the operation of the device by a complete mathematical model.

10. Major publications of Research Results:

(a) *Contribution to International Journals and refereed proceedings*

1. MA Islam, R Ahmed, SMY Bhuiyan, "On the Cycle Frequency of a Thermoloop Device," to be submitted in Int. J. Heat and Mass Transfer.
2. MA Islam, R Ahmed, SMY Bhuiyan, "Operating Characteristics of a Pulsated Two-Phase Loop Thermosyphon," to be submitted Applied Thermal Engineering.

(b) *Contribution to Conferences / Seminars*

3. K Sharmin and MA Islam, "Pressure and Operating Characteristics of a Pulsated Two-Phase Loop Thermosyphon," to be presented and included in the *Proc. ICME2009* to be held during 26-28 December 2009, Dhaka, Bangladesh.
4. MA Islam, R Ahmed, SMY Bhuiyan, "Effect of Working Fluids and Evaporator Geometry on the Performance of Thermoloop," *Proc. 4th BSME-ASME International Conference on Thermal Engineering*, 27-29 December, 2008, Dhaka, Bangladesh
5. MA Islam, MA Rahman, M Alam, "Heat Transfer Performance of a Pulsated Two-Phase Loop Thermosyphon," *Proc. IMECE2008*, November 2-6, 2008, Boston, Massachusetts, USA

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